Natural Draft Cooling Tower Performance Evaluation

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Abstract:

Cooling towers are used extensively for numerous, residential, commercial and industrial applications. The heat rejected and water evaporated in natural draft cooling towers are critically evaluated by employing the Merkel and e–number-of-transfer-units (e-NTU) methods of analysis, respectively, at different operating and ambient conditions.

The importance of using a particular method of analysis when evaluating the performance characteristics of a certain fill material and subsequently employing the same analytical approach to predict cooling tower performance is stressed. The effect of ambient humidity and temperature on the performance of cooling towers employing the Merkel and e-NTU methods of analysis are evaluated.

1. Introduction:

Cooling towers are used widely in industrial processes for releasing the waste heat arising into environment. A cooling tower in HVAC application is widely used. Several types of cooling tower is available, the counter flow and forced draft cross flow are most commonly ones used in HVAC applications.

The cooling tower is a device which reticulating cooling water from heat exchangers is cooled by contact with atmospheric air. The air coming from the air inlet will interact with the smaller droplets of warm water which flow down the porous media where heat transfer will occur. The cooled down water droplets will then flow down to be collected at the bottom of the cooling tower to be recirculated back into the system and then the cycle continues [1].

Walker et al. [2] was suggest the operation theory of cooling tower for the first time. The theory of cooling towers has been studied in some depth since the first work of Merkel in 1925 [3]. It is a reasonably accurate and relatively simple mathematical description of the heat and mass transfer phenomena in a counter current tower. It was the first to present the practical use of basic differential equations, in which he combined the equations for heat and water vapor transfer. The basic assumptions that are inherent in Merkel's theory are:

- The resistance for heat transfer in the liquid film is negligible;
- The mass flow rate of water per unit cross sectional area of the tower is constant, i.e. there is no loss of water due to evaporation;
- The specific heat of the air–steam mixture at constant pressure is the same as that of dry air;
- The Lewis number for humid air is unity.

It should be noted that the formulation and implementation of Merkels theory in cooling tower design and performance evaluation is presented and discussed in most unit operations and process heat transfer textbooks. This assumption has been generally accepted in theoretical analyses and cooling tower design.

Classification of cooling tower

Most cooling towers used in commercial refrigeration plants for or industrial buildings applications are mechanical draft cooling towers uses fans to extract atmospheric air. A cooling tower consists of a fan to extract intake air, a heat transfer medium or packing, a water basin, a water distribution system, and an outer casing. According to the location of the fan corresponding to the packing and to the flow arrangements of air and water, current widely used mechanical draft cooling tower for HVAC and industrial applications can be classified into the following categories [4]:

1-Natural-circulation cooling towers,
2-Counter flow induced draft cooling towers,
3-Cross flow induced draft cooling towers; and

4-Counter flow forced draft cooling towers.

Heat is discharged in power generation, refrigeration, petrochemical, steel, processing and many other industrial plants. In many cases, this heat is discharged into the atmosphere with the aid of a cooling tower. Fig. (1) shows an example of the application of a cooling tower in a simple steam power plant. Heat is discharged into the atmosphere by the cooling tower via a secondary cycle with water as the process fluid.

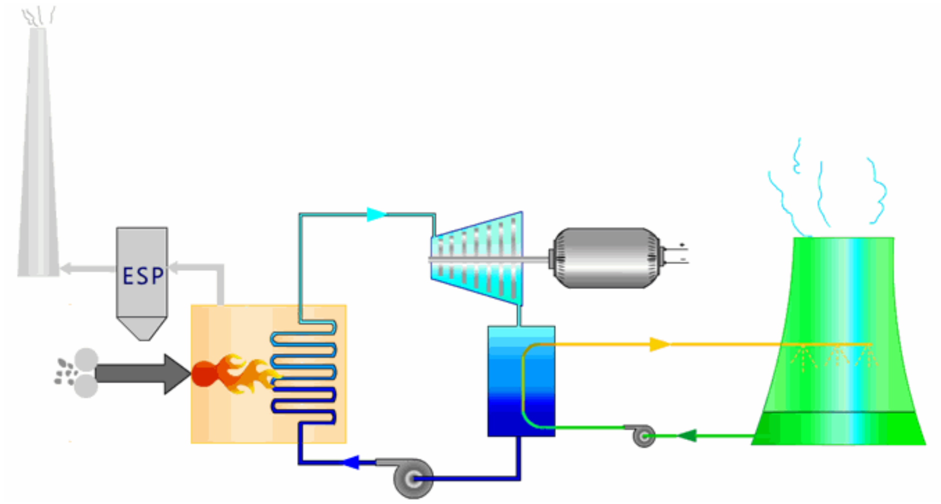

Fig. (1) Simple steam Power plant with cooling tower

Wet-cooling towers are considered in this study. Wet-cooling takes place when the water is in direct contact with the air. Cooling is the result of sensible and latent heat transfer where the latent heat transfer component generally dominates.

2. Review of the previous work:

Fisenko, et al [5] was developed a mathematical model of control system for the mechanical draft cooling tower, the model discussed the heat and mass transfer processes between water film and turbulent draft air flow at quasi state approximation. Different regimes of cooling tower performance are compared and the optimization method is proposed too. J. Khan, et al [6] studied a comprehensive design and performance evaluation study of counter flow wet cooling towers. It was found that, the sensitivity of the effectiveness and water outlet temperature with respect to inlet air wet bulb temperature and water inlet temperature is investigated for different (L/G) mass flow rate ratios.

Jorge Facao [7] focused on computational data analysis of heat and mass transfer in an indirect contact cooling tower. The computational fluid dynamics (CFD) model uses as boundary conditions the temperatures of the tubes obtained by a correlation model. The available mass transfer correlations for indirect cooling towers presented and compared with a correlation obtained from(CFD) simulations.

Webb [8] performed a unified theoretical treatment for thermal analysis of cooling towers, evaporative condensers and evaporative fluid coolers. Specific calculation procedures are explained for sizing and rating each type of evaporative exchanger. Webb and Villacres [9] described three computer algorithms that have been developed to perform rating calculations of three evaporatively cooled heat exchangers. At part load conditions, the algorithms are particularly useful for rating commercially available heat exchangers. The heat and mass transfer ‘‘characteristic equation’’ of one of the heat exchangers is derived from the manufacturers rating data at the design point.

Braun et al. [10] presented the cooling towers and cooling coils effectiveness models. These models utilize existing thermal effectiveness relationships developed for sensible heat exchangers with modified definitions for the number of transfer units and the fluid capacitance rate ratio. The results of the models were compared with those of more detailed numerical solutions to the basic heat and mass transfer equations and experimental data. They also did not consider the effect of air–water interface temperature, however, they did consider the effect of water evaporation on the air process states along the vertical length of the tower. The results are displayed only for a Lewis number equal to unity.

Nimr [11] presented a mathematical model to describe the thermal behavior of cooling towers that contain packing materials. The model takes into account both sensible and latent effects on the tower performance. A closed form solution was obtained for both the transient and steady temperature distribution in a cooling tower. De Villiers and Kroger [12], developed relations for various geometries and configurations and explained that the mass transfer relation could be calculate an effective drop diameter, a diameter that would have the same effect as the actual band of drops in the tower. Kloppers and Kroger [13], investigated the effect of the Lewis factor, or Lewis relation, on the performance prediction of natural draft and mechanical draft cooling towers. They found that if the same definition of Lewis factor is employed in the fill test analysis and in the subsequent cooling tower performance analysis, the water outlet temperature would be accurately predicted.

Kloppers and Kroger [14] have proposed and discussed many other mathematical models which correlated heat and mass transfer processes occurring in wet cooling towers. Bilal A. Qureshi , [15] predicted that evaporation losses is significant because water in cooling towers is cooled primarily through the evaporation of the part of the circulating water, which causes the concentration of dissolved solids and other impurities to increase .The predicted values are in good agreement with experimental data as well as predictions made by an accurate mathematical model.

Lijuan [16]developed a new model based on the double film theory for air-cooling towers thermodynamic calculation. Rafat [17],Investigated numerically the effect of wind break walls on the thermal performance of natural draft wet cooling tower (NDWCT) under crosswind.

Poppe and Rogener [18], developed a new model for cooling towers that were not use the simplifying assumptions made by Merkel, in their study different packing are studying. Khan et al.[19], presented mathematical modeling of cooling towers integrate fouling growth model , in addition to considering effect of pressure and fouling on thermal cooling tower performance. Karami and Heidarinejad [20], developed heat and mass transfer characteristic of wet counter-flow cooling tower. They presented by increasing in mass flow ratio, tower effectiveness is increased but temperature ratio is decreased.

3. Governing equations for heat and mass transfer in fill for unsaturated air:

Merkel's theory:

Fig (2) shows a control volume in the fill of a counter flow wet cooling tower and Fig. (3) shows an airside control volume of the fill which shown in Fig. (2).
A mass balance for the control volume in Fig.(2) yields,

$$dm_w = m_a dw \tag{1}$$

The energy balance for the control volume in Fig.(2) is as follows:

$$m_a di_{ma} = m_a di_w + i_w dm_w \tag{2}$$

Where i_{ma} is the enthalpy of the air vapor mixture which expressed by the following equation:

$$i_{ma} = c_{pa}(T - 273.15) + w[i_{fgwo} + c_{pv}(T - 273.15)] \quad J/kg\, air\, vapor \tag{3}$$

where the latent heat (i_{fgwo}) is calculated from the following equation:

$$\begin{aligned} i_{fgw} &= 3.4831814\times10^{6} - 5.8627703\times10^{3}T + 12.139568T^{2} \\ &\quad -1.40290431\times10^{-2}T^{3} \quad J/K \end{aligned} \tag{4}$$

substitute equation (1) into equation (2) and rearrangement as follows:

$$dT_w = \frac{m_a}{m_w}\left(\frac{1}{c_{pw}} di_{ma} - T_w dw\right) \tag{5}$$

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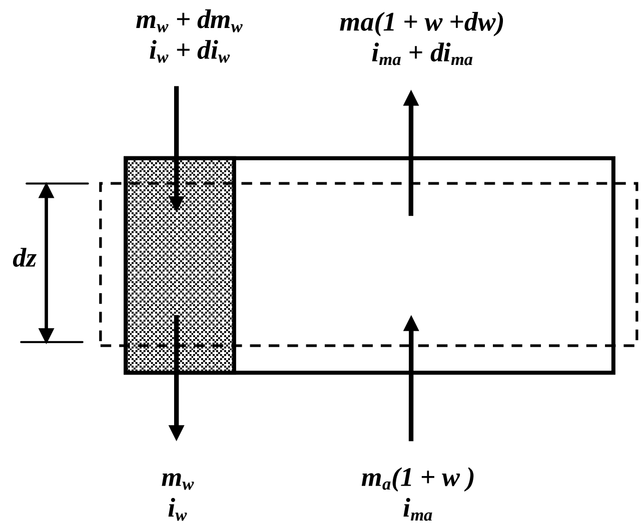

Fig. (2) Control Volume of the Counter Flow Fill

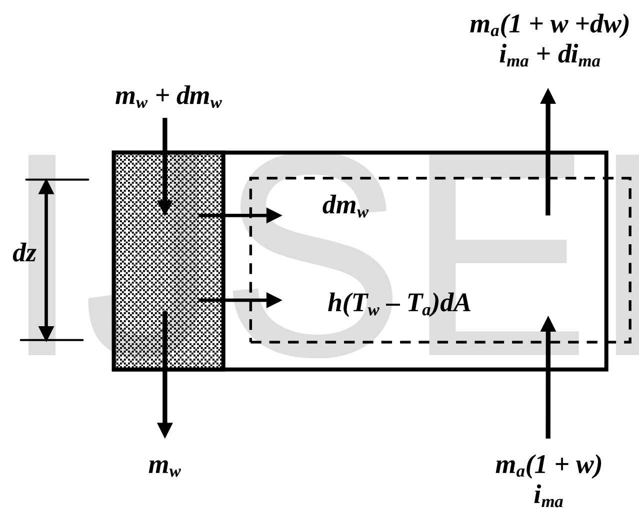

Fig. (3) Air Side Control Volume of the Fill

Consider the interface between the air and the water in Fig. (2). The energy balance for the control volume as follows:

$$dQ = dQ_m + dQ_c \tag{6}$$

Where dQ_m is the enthalpy transfer due to difference in vapor concentration between the saturated air at the interface and the mean stream air and dQ_c is the sensible heat transfer due to the difference in temperature. The mass transfer at the interface is expressed by,

$$dm_w = h_d\,(w_{sw} - w)dA \tag{7}$$

The corresponding enthalpy transfer for the mass transfer in equation (7) is

$$dQ_m = i_v dm_w = i_v h_d\,(w_{sw} - w)dA \tag{8}$$

The enthalpy of the water vapor, i_v at the bulk water temperature, T_w is given by:

$$i_v = i_{fgwo} + c_{pv} T_w \tag{9}$$

The convective heat transfer from Fig. (3) is given by:

$$dQ_c = h(T_w - T_a) dA \tag{10}$$

The temperature differential in equation (10) can be substituted by an enthalpy differential. The enthalpy of saturated air evaluated at the local bulk water temperature is given by:

$$i_{masw} = c_{pa} T_w + w_{sw}(i_{fgwo} + c_{pv} T_w) \tag{11}$$

Substitute equation (9) into equation (11) and rearrangement

$$i_{masw} = c_{pa} T_w + w i_v + (w_{sw} - w) i_v \tag{12}$$

The enthalpy of the air-water vapor mixture per unit mass of dry air, which according to the equation (3), where the specific heat are evaluated at (T+273.15)/2 and the latent heat i_{fgwo} is evaluated at 273.15 K according to equation (4) is expressed by:

$$i_{ma} = c_{pa} T_a + w(i_{fgwo} + c_{pa} T_a) \tag{13}$$

The specific heat of the air-water vapor mixture fur unsaturated air is given by

$$c_{pma} = c_{pa} + w c_{pv} \tag{14}$$

Subtract equation (13) from (12). The resultant equation can be simplified if the small differences in specific heats, which are evaluated at different temperatures, are ignored.

$$T_w - T_a = \frac{(i_{masw} - i_{ma}) - (w_{sw} - w) i_v}{c_{pma}} \tag{15}$$

where c_{pma} is given by equation (14).

Substitute equation (15) into equation (10). Substitute the resultant equation and equation (8) into equation (6) and rearrangement,

$$dQ = h_d \left(\frac{h}{c_{pma} h_d} (i_{masw} - i_{ma}) + \left[1 - \frac{h}{c_{pma} h_d} \right] i_v (w_{sw} - w) \right) dA \tag{16}$$

$\frac{h}{c_{pma} h_d}$ is known as the Lewis factor, Le_f, and is an indication of the relative rates of heat and mass transfer in an evaporative process. Bosnjakovic [21] developed an empirical relation for the Lewis factor, Le_f, for air-water vapor systems. The Lewis factor for unsaturated air, according to Bosnjakovic [21] is given by

$$Le_f = 0.865^{0.667} \frac{\left(\frac{w_{sw} + 0.622}{w + 0.662} - 1 \right)}{\ln\left(\frac{w_{sw} + 0.622}{w + 0.622} \right)} \tag{17}$$

The enthalpy transfer to the air stream from equation (16) is

$$di_{ma} = \frac{dQ}{m_a} = \frac{h_d dA}{m_a}\left[Le_f(i_{masw} - i_{ma}) + (1 - Le_f)i_v(w_{sw} - w)\right] \tag{18}$$

For a one-dimensional model of the cooling tower fill, where the available area for heat and mass transfer is the same at any horizontal section through the fill, the transfer area for a section dz is usually expressed as

$$dA = a_{fi} A_{fr} dz \tag{19}$$

where a_{fi} is the area density of the fill, i.e. the wetted area divided by the corresponding volume of the fill and A_{fr} is the corresponding frontal area or face area. Substitute equation (19) into equation (18), get

$$\frac{di_{ma}}{dz} = \frac{h_d a_{fi} A_{fr}}{m_a}\left[Le_f(i_{masw} - i_{ma}) + (1 - Le_f)i_v(w_{sw} - w)\right] \tag{20}$$

To simplify the analysis of an evaporative process Merkel [3] assumed that the evaporative loss is negligible, i.e. dw = 0 from equation (5), and that the Lewis factor is equal to unity. The governing equations (20) and (5) of the counter flow evaporative process simplify respectively to

$$\frac{di_{ma}}{dz} = \frac{h_d a_{fi} A_{fr}}{m_a}(i_{masw} - i_{ma}) \tag{21}$$

and by dividing equation (5) by dz on both sides of equation (5) to

$$\frac{dT_w}{dz} = \frac{m_a di_{ma}}{m_w c_{pw} dz} \tag{22}$$

Equations (21) and (22) describe respectively the change in the enthalpy of the air-water vapor mixture and the change in water temperature as the air travel distance changes. Equations (21) and (22) can be combined to yield upon integration the Merkel equation,

$$Me_M = \frac{h_d A}{m_w} = \frac{h_d a_{fi} A_{fr} L_{fi}}{m_w} = \frac{h_d a_{fi} L_{fi}}{G_w} = \int_{t_{wo}}^{t_{wi}} \frac{c_{pw} dT_w}{(i_{masw} - i_{ma})} \tag{23}$$

Where Me_M is the Merkel number according to the Merkel approach. It is not possible to calculate the state of the air leaving the fill according to equation (23). Merkel assumed that the air leaving the fill is saturated with water vapor. This assumption enables the air temperature leaving the fill to be calculated.

e-NTU Method:

Jaber and Webb [22] developed the equations necessary to apply the e-NTU method directly to counter flow or cross flow cooling towers. It can be shown according to Jaber and Webb [22] that

$$\frac{d(i_{masw} - i_{ma})}{(i_{masw} - i_{ma})} = h_d\left(\frac{di_{masw}/dT_w}{m_w c_{pw}} - \frac{1}{m_a}\right)dA \tag{24}$$

Equation (24) corresponds to the heat exchanger e-NTU equation

$$\frac{d(T_h - T_c)}{(T_h - T_c)} = -U\left(\frac{1}{m_h c_{ph}} + \frac{1}{m_c c_{pc}}\right)dA \tag{25}$$

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Two possible cases of equation (24) can be considered where rn$_a$ is greater or less than $m_w c_{pw} /(di_{masw} / dT_w)$. The maximum of m$_a$ and $m_w c_{pw} /(di_{masw} / dT_w)$ is denoted by C_{max} and the minimum by C_{min}. The gradient of the saturated air enthalpy-temperature curve is

$$\frac{di_{masw}}{dT_w} = \frac{i_{maswi} - i_{maswo}}{T_{wi} - T_{wo}} \tag{26}$$

The fluid capacity rate ratio is defined as

$$C = C_{\min} / C_{\max} \tag{27}$$

The effectiveness is given by

$$e = \frac{Q}{Q_{\max}} = \frac{m_w c_{pw} (T_{wi} - T_{wo})}{C_{\min} (i_{maswi} - \lambda - i_{mai})} \tag{28}$$

Where λ. is a correction factor, according to Berman [23], to improve the approximation of the i_{masw} versus T_w curve as a straight line. The correction factor, λ is given by

$$\lambda = (i_{maswo} + i_{maswi} - 2i_{maswm}) / 4 \tag{29}$$

where i_{maswm} donates the enthalpy of saturated air at the mean water temperature. The transfer units number for counter flow cooling towers is given by

$$NTU = \frac{\ln \dfrac{1 - eC}{1 - e}}{1 - C} \tag{30}$$

If m$_a$ is greater than $m_w c_{pw} /(di_{masw} / dT_w)$ the Merkel number according to the e-NTU approach is given by

$$Me_e = \frac{c_{pw}}{di_{masw} / dT_w} NTU \tag{31}$$

If m$_a$ is less than $m_w c_{pw} /(di_{masw} / dT_w)$ the Merkel number according to the e-NTU approach is given by

$$Me_e = \frac{m_a}{m_v} NTU \tag{32}$$

The performance of natural draft counter flow cooling towers is evaluated by respectively employing the Merkel and e-NTU methods of analysis at different operating and ambient conditions. The importance of using a particular method of analysis when evaluating the performance characteristics of a certain fill material and subsequently employing the same analytical approach to predict cooling tower performance, is investigated.

Fig. (4) to Fig. (8), illustrate respectively the heat rejected, Q, the water outlet temperature, T_{wo}. the air outlet temperature, T_{ao}, the mean air-water vapor mass flow rate, m_{av}, and the mass flow rate of the water evaporated from the water stream, $m_{w(evap)}$ the inlet air is varied from dry to saturated conditions where the ambient temperatures are equal to 280, 290 and 300 K. The solid line in each of the figures represents the results according to the e-NTU approach while the broken lines represent the results according to the more rigorous Merkel approach.

Heat Rejected

The heat rejected by the cooling tower at ambient temperatures of 280, 290 and 300 K in dry to saturated conditions, as shown in Fig. (4). It can be notice that the heat rejection predicted by e-NTU approach is higher than that predicted by the Merkel approach at all the ambient conditions considered in this investigation. The e-NTU approach predicts higher heat rejection rates Q than the Merkel approach as shown in Fig. (4). This is because the Merkel approach ignores the loss in water mass flow rate in the energy equation. It is evident that the difference in heat rejection rates between the Merkel and e-NTU approaches increases as the inlet air becomes dryer and hotter.

Water Outlet Temperature

The water outlet temperature at ambient temperatures of 280,290 and 300 K in dry to saturated conditions, is shown in Fig. (5). The water outlet temperatures predicted by the Merkel and e-NTU approaches are practically identical where the draft through the tower is approximately the same as shown in Fig. (5). The Merkel numbers, determined by the e-NTU approach for the expanded metal fill employed in this natural draft cooling tower analysis, is approximately 1% lower than the Merkel number determined by the Merkel approach.

It is expected that the results of the Merkel and e-NTU approaches must be identical since the same simplifying assumptions are used in these methods. However, Fig. (5) shows the differences in the predicted performance by the Merkel and e-NTU approaches. The reason why the predicted performance is not the same for both approaches is because the cooling tower fill was originally tested at different ambient and operating conditions than where it was subsequently applied in this investigation.

Air Outlet Temperature

The air outlet temperatures predicted by the e-NTU approach are higher than those predicted by the Merkel approach in all the ambient conditions considered is illustrated in Fig. (6). When the ambient temperature is low, the discrepancy between the predicted air outlet temperatures is the smallest. When the temperature of the ambient air increases, the discrepancy between the predicted air outlet temperatures increases in very dry conditions. When the humidity increases at a given temperature, the discrepancy decreases.

The draft through natural draft cooling towers is a function of the density of the air above the fill. It is thus very important to predict the air temperature above the fill accurately. The Merkel and e-NTU methods are unable to predict the temperature of the outlet air without the assumption that the outlet air is saturated with water vapor.

Cooling tower air outlet temperatures generally increase when air inlet temperatures and humidity increase, as can be seen in Fig. (6). In very hot very dry conditions the air outlet temperature can be less than the air inlet temperature.

Mean Air-Water Vapor Mass Flow Rate (Tower Draft)

The mean air-water vapor mass flow rates, determined by the e-NTU approach, are higher than those predicted by the Merkel approach at all the ambient conditions considered is shown in Fig. (7). The mean air-water vapor mass flow rate is strongly coupled to the air outlet temperature. This is because the density of the air inside the cooling tower is a function of the air temperature. The mass flow rate of air through the tower is, in turn, a function of the density differential of the air internal and external to the cooling tower. Thus, the draft through the natural draft cooling tower is strongly coupled to the air outlet temperature. The draft, in turn, will influence the heat rejection rate in the cooling tower. It is clear that the processes in a natural draft cooling tower are strongly coupled. At temperatures of 280, 290 and 300 K, the air-vapor mass flow rates increase as the inlet ambient humidity ratio is increased.

For natural draft towers, however, the discrepancy between the Merkel and e-NTU approaches increases as the air gets warmer and drier. This is because the air outlet temperature (T_{ao}) and tower draft or air-water vapor mass flow rate (m_{av}) are strongly coupled for natural draft towers. When compared to the case where the ambient temperature is 300 K, it can be seen that the discrepancy is large between the Merkel and e-NTU approaches for the draft and air outlet temperatures because of the higher draft of the e-NTU method at higher air temperatures with lower humidity, more cooling is taking place.

Water Evaporation Rate

The predicted water evaporation rates in natural draft cooling towers are always higher according to the e-NTU approach than according to the Merkel approach as shown in Fig. (8). This is the case even if the outlet air is unsaturated, according to the e-NTU approach. The air can be unsaturated, according to the e-NTU approach, but the predicted evaporation rate is still higher than that predicted by the Merkel approach where the outlet air is saturated, because of the strongly coupled draft and energy equations. The hotter the air, the higher the draft. The higher the draft, the more heat and mass transfer and thus higher evaporation rates.

4. Conclusion

The heat rejected and water evaporated in natural draft cooling towers are critically evaluated by employing the Merkel and e–number-of-transfer-units (e-NTU) methods of analysis, respectively, at different operating and ambient conditions. The predicted water evaporation rates in natural draft cooling towers are always higher according to the e-NTU approach than according to the Merkel approach. The e-NTU and Merkel approaches predict virtually the same tower performances when the models are applied consistently or inconsistently.

The heat transfer rate, water outlet temperature, draft, air outlet temperature and evaporation rate of the Merkel approach can be brought within closer tolerances of the more rigorous e-NTU approach, when the reduction of the water mass flow rate, due to evaporation, is included in the energy balance. The assumption of Merkel that the outlet air is saturated with water vapor, leads to tower performance that are within close tolerance of the tower performance predicted by the e-NTU approach, for cold or humid ambient conditions.

LIST OF SYMBOLS

A	Area, m^2
a	Surface area per unit volume, m^{-1}, or coefficient
C	Fluid capacity rate kg/s ,C_{min}/C_{max}
c	Concentration, kg/m^3, or constant
c_p	Specific heat at constant pressure, J/kgK
e	Effectiveness
G	Mass velocity, kg/m^2s
h	Heat transfer coefficient, W/m^2K, or equality constraint function
h_d	Mass transfer coefficient, kg/m^2s
i	Enthalpy, J/kg
i_{masw}	Enthalpy of saturated air at the local bulk water temperature, J/kg
i_{fg}	Latent heat, J/kg
J	Momentum flux, kg m/s^2
L	Length, m
l	Characteristic length
M	Molecular weight, kg/mole
m	Mass flow rate, kg/s
Me	Merkel number
NTU	Number of transfer units
Q	Heat transfer rate, W
T	Temperature, °C or K
U	Overall heat-transfer coefficient, W/m^2 K
w	Humidity ratio, kg water vapor/kg dry air
w_{sa}	Humidity ratio of saturated air at T_a , kg/kg
w_{sw}	Saturation humidity ratio of air evaluated at the local bulk water temperature, kg/kg
z	Coordinate, or elevation, m, or exponent

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Subscripts:

a	Air, or above
d	Drop, or discharge, or day
e	Evaporative, or expansion, or e-NTU theory
fi	Fill
fr	Frontal
i	Inlet
M	Merkel theory
m	Mean, or mass transfer
max	Maximum
min	Minimum
o	Outlet
s	Saturation, or shell
ss	Supersaturated
v	Vapor
w	Water

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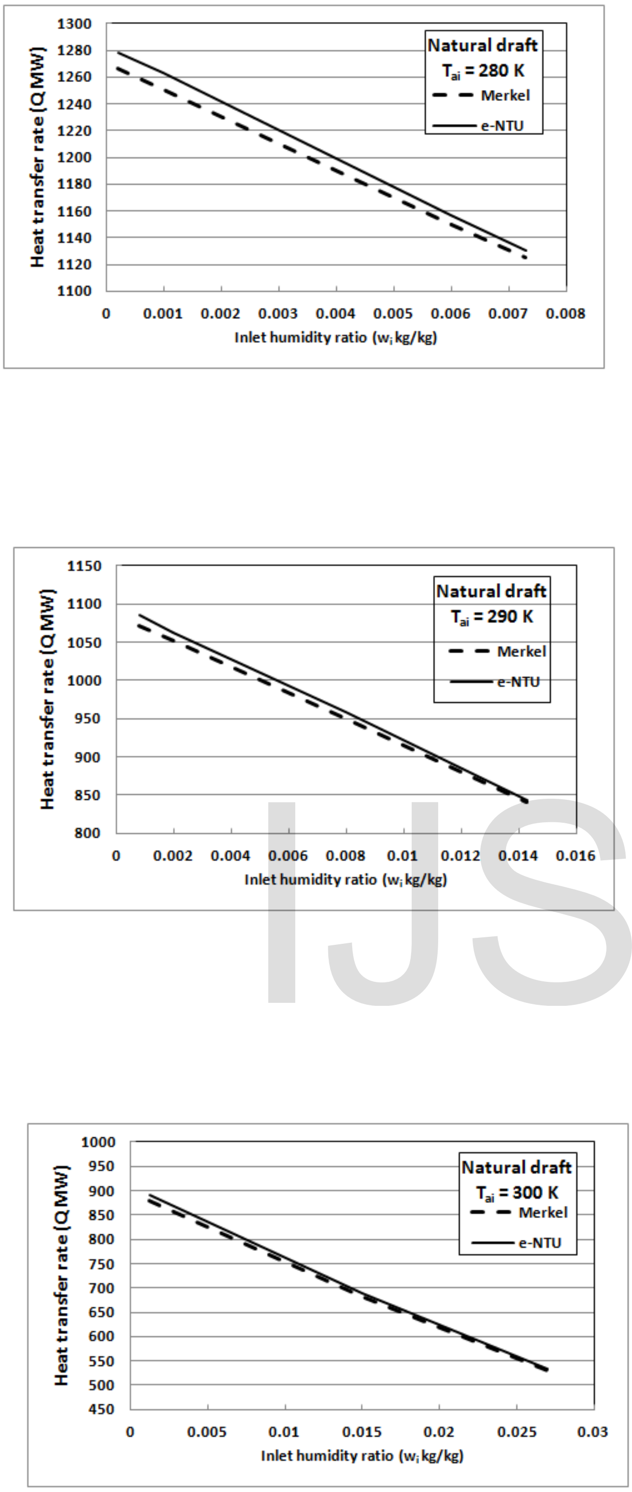


Fig. (4) Heat rejected versus inlet humidity ratio

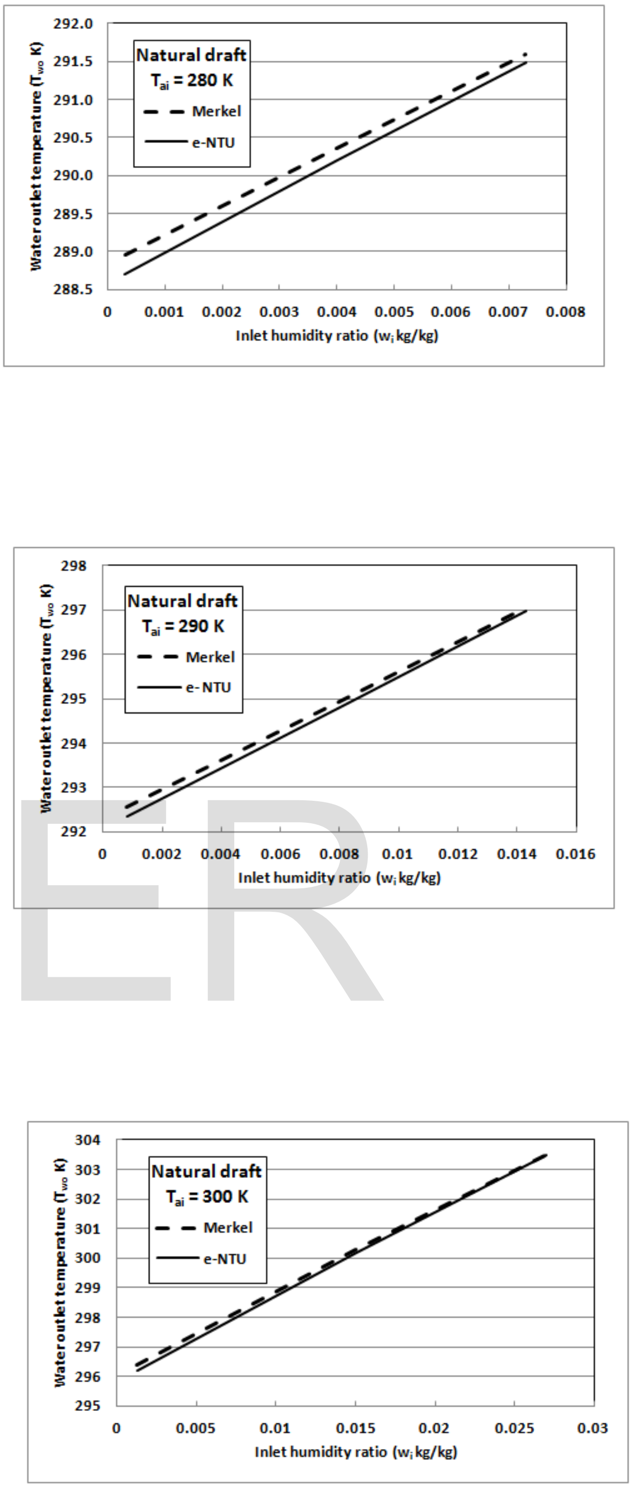
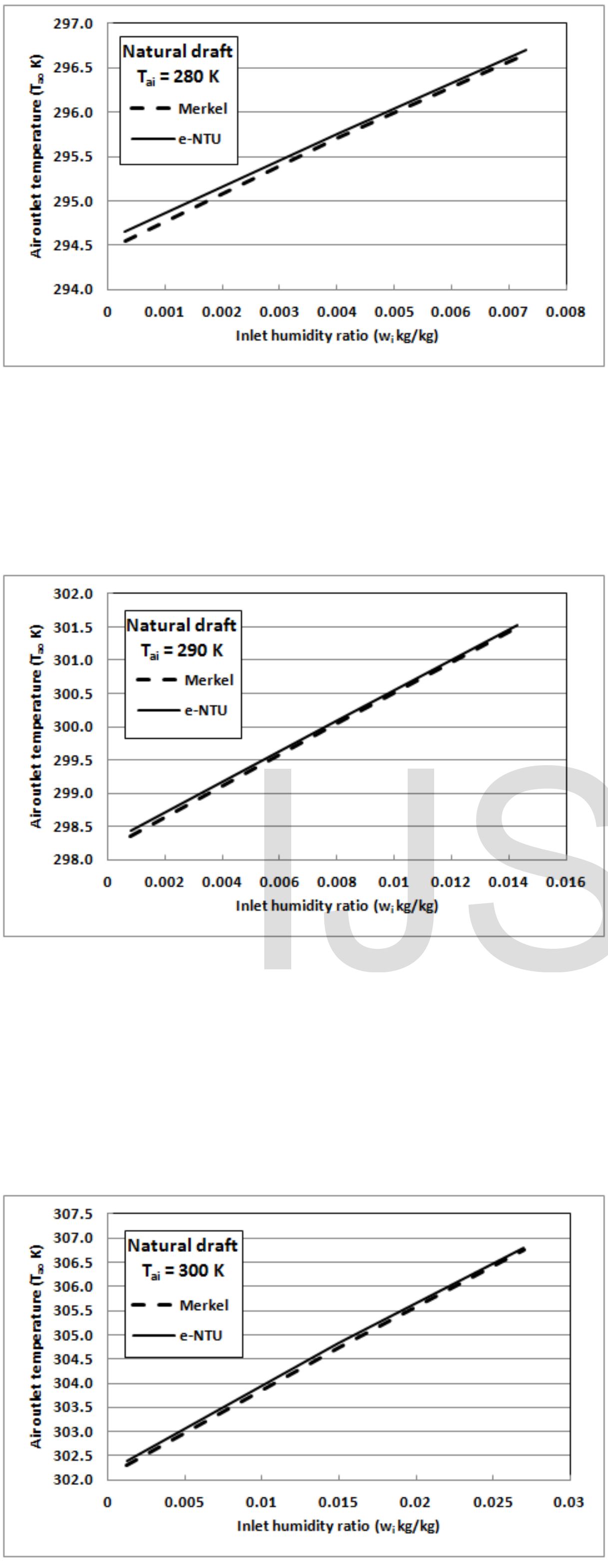


Fig. (5) Water outlet temperature versus inlet humidity ratio



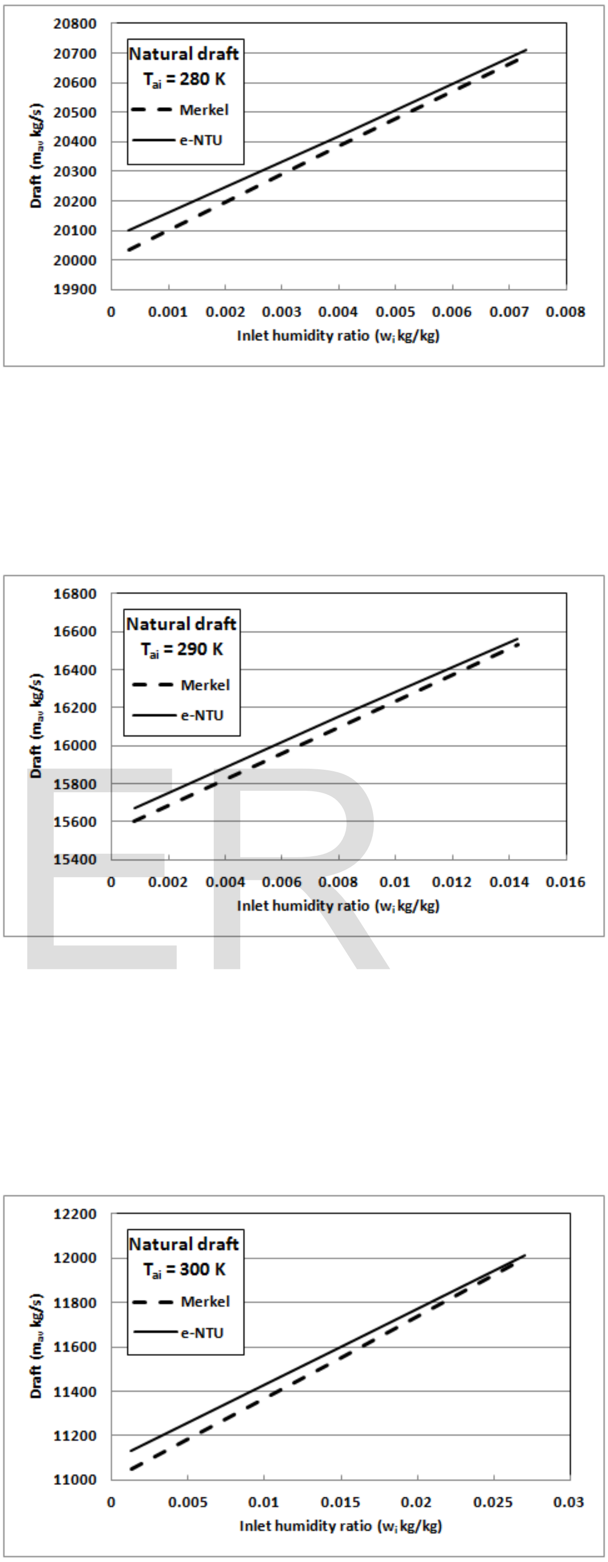
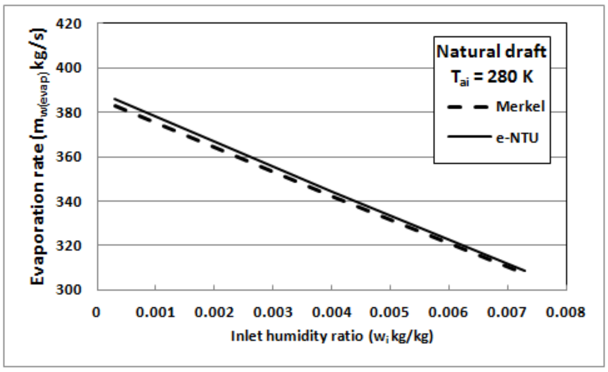
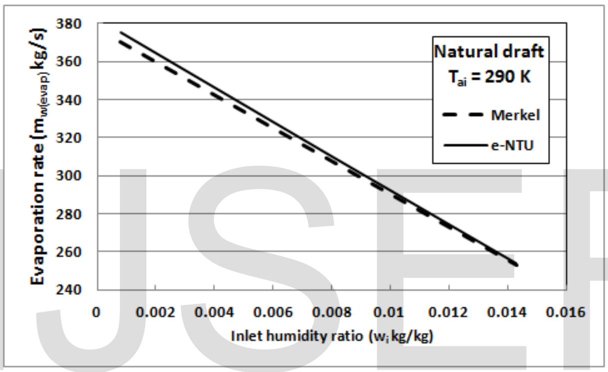


Fig. (6) Air outlet temperature versus inlet humidity ratio

Fig. (7) Draft versus inlet humidity ratio





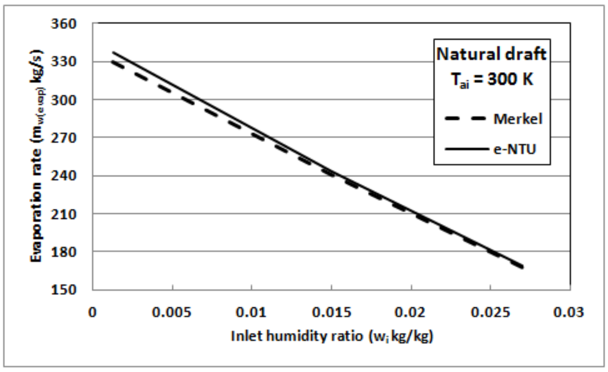


Fig. (8) Evaporation rate versus inlet humidity ratio