ENERGY AND EXERGY ANALYSIS OF COUNTER FLOW WET COOLING TOWERS

by

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Cooling tower is an open system direct contact heat exchanger, where it cools water by both convection and evaporation. In this paper, a mathematical model based on heat and mass transfer principle is developed to find the outlet condition of water and air. The model is solved using iterative method. Energy and exergy analysis infers that inlet air wet bulb temperature is found to be the most important parameter than inlet water temperature and also variation in dead state properties does not affect the performance of wet cooling tower.

Key words: *rating, energy, exergy, cooling tower, dead state, second law*

Introduction

Cooling towers are basically an open system direct contact heat exchanger, where it is used thermally to reclaim circulating water for reuse in power plant condensers, refrigerant condensers and other heat exchangers. The warm water is admitted at the top of the tower and moves counter flow to the air. Waste heat present in the warm water is rejected to the atmospheric air through convection and evaporation heat transfer. Merkel [1] developed a mathematical model for cooling towers using differential equations. In his work, sensible and latent heat transfer processes occurring in the tower are combined into a single process, based on enthalpy difference as the driving potential, termed the total heat transfer process. In this the water loss by evaporation is neglected. Threlkeld [2] analysed the cooling tower, taking into consideration, the water loss due to evaporation and the actual Lewis number, unlike the assumptions made in Merkel's model. It is reported [3] that Merkel's model underestimates the tower volume by 5-15% depending on the operating parameters. Zubair *et al*. [4] presented a detail model of counter flow wet cooling towers and showed that a majority mode of heat transfer rate is evaporation. Kloppers *et al*. [5] proposed the influence of Lewis number on the performance of wet cooling towers.

Usage of exergy concepts in evaluating the performance of energy systems are increasing nowadays due to its clear indication of loss at various locations which is more informative than energy analysis. Exergy is the work potential of energy in a given environment [6]. Rosen and Dincer [7] studied the effect of dead state variation on energy and exergy analysis of thermal systems and showed that the variation does not affect the energy and exergy values significantly. In exergy analysis losses are measured in terms of exergy destruction, which provide direct measure of thermodynamic inefficiencies. Oman *et al.* [8] studied this, through the experiment with natural draft cooling tower. References [9-13] contain detailed view about exergy and its use in various applications. Moran [10] discussed the exergy analysis of cooling tower through an example problem. Qureshi *et al*. [14] carried out second law analysis of cooling tower and evaporative heat exchanger, showed that process taking place in these devices approaching reversible. Wanchai *et al*. [15] developed a mathematical model with respect to tower height and exergy analysis. From the results showed that exergy destruction (entropy generation) is more at the bottom of the tower and least at the top for the conditions considered. Inlet air dry bulb temperature has insignificant effect on wet cooling tower performance for the same tower configuration [16].

The objective of this paper is to theoretically study the heat and mass transfer characteristics of counter flow wet cooling tower. Rating, energy, and exergy analysis based on developed mathematical model is carried out. Effect of dead state on second law efficiency is also studied.

Mathematical model

Heat and mass transfer characteristics of the evaporative cooling tower can be determined by mass and energy balance. The mathematical model is developed with the main following assumptions [2, 4]:

- heat and mass transfer is in a direction normal to the flows only,
- negligible heat and mass transfer through the tower walls to the environment,
- negligible heat transfer from the tower fans to the air or water streams,
- constant water and dry air specific heats,
- constant heat and mass transfer coefficients throughout the tower,
- water lost by drift is negligible,
- uniform temperature throughout the water stream at each cross-section,
- uniform cross-sectional area of the tower, and
- the Lewis number for humid air is unity.

By considering the control volume of each segment as shown in fig. 1 the energy balance can be written as follows:

$$G\mathrm{d}h = L\mathrm{d}h_{\mathrm{f\,w}} + G\mathrm{d}Wh_{\mathrm{f\,w}} \tag{1}$$

- water energy balance:

$$L\mathrm{d}h_{\mathrm{f\,w}} = h_{\mathrm{c}}A_{\mathrm{V}}\mathrm{d}V(t_{\mathrm{w}} - t) + h_{\mathrm{d}}A_{\mathrm{V}}\mathrm{d}V(W_{\mathrm{s\,w}} - W)h_{\mathrm{fg\,w}} \tag{2}$$

- airside water vapor mass balance:

$$G\mathrm{d}W = h_{\mathrm{d}}A_{\mathrm{V}}(W_{\mathrm{s\,w}} - W) \tag{3}$$

Substituting Lewis number into eq. (2), gives:

$$L\mathrm{d}h_{\mathrm{f\,w}} = h_{\mathrm{d}}A_{\mathrm{V}}\mathrm{d}V[\mathrm{Le}C_{\mathrm{pa}}(t_{\mathrm{w}} - t) + (W_{\mathrm{s\,w}} - W)h_{\mathrm{fg\,w}}] \tag{4}$$

From eqs. (1) and (3):

$$L\mathrm{d}h_{\mathrm{f\,w}} = G\mathrm{d}h - h_{\mathrm{d}}A_{\mathrm{V}}(W_{\mathrm{s\,w}} - W)h_{\mathrm{f\,w}} \tag{5}$$

Combining eqs. (5) and (3) , we get:

$$\frac{\mathrm{d}h}{\mathrm{d}W} \quad \mathrm{Le}C_{\mathrm{pa}}\frac{(t_{\mathrm{w}} \quad t)}{(W_{\mathrm{s\,w}} \quad W)} \quad h_{\mathrm{g\,w}} \tag{6}$$

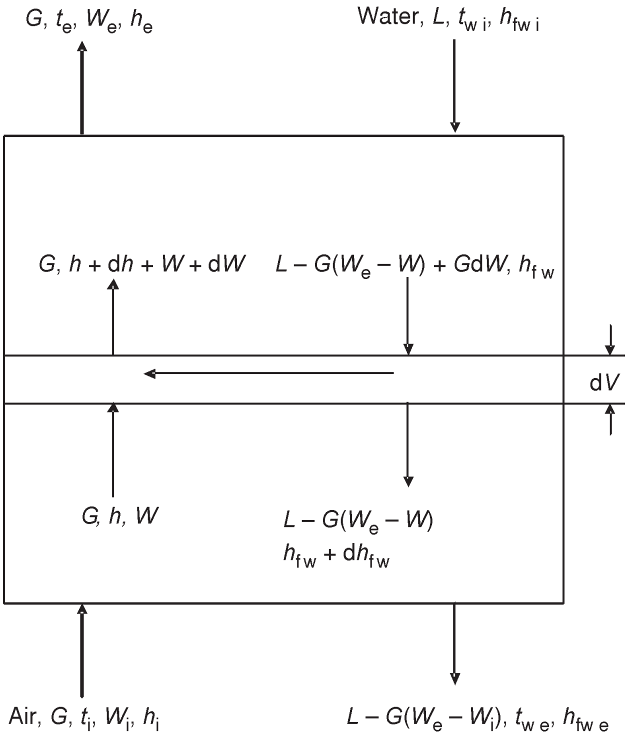

Figure 1. Mass and energy balance of counter flow wet cooling tower

Using the approximation of constant C_{pa} , we have:

$$h_{sw} \quad h \quad C_{pa}(t_{w} \quad t) \quad h_{g}^{0}(W_{sw} \quad W) \tag{7}$$

Equation (6) may then written as:

$$\frac{dh}{dW} \quad \mathrm{Le}\frac{(h_{sw} \quad h)}{(W_{sw} \quad W)} \quad h_{gw} \quad \mathrm{Le}h_{g}^{0} \tag{8}$$

Eqution (6) describes the condition line on the psychometric chart for the changes in state for moist air passing through the tower. In this regard, air-water vapor thermodynamic properties are calculated by equations based on ASHRAE [17]. For given inlet condition of air, mass flow rates (air, water), and water temperatures eqs. (1) and (8) are solved simultaneously using iterative method to find the exit conditions of both air and water stream. Since it is evaporative cooling, water flow rate along the height of the tower is varying. Thus, water flow rate at the bottom of the tower is unknown. To start the iteration, an initial guess is made in such a way that it should match the inlet condition. The model is validated using experimental values reported in Simpson and Sherwood [18]. The error percentage in predicted and experimental values of outlet air wet bulb temperature ($t_{wb\,e}$) is within 0.5%.

Analytical review

The number of transfer units (NTU) representing the size of the cooling towers can be calculated from [4]:

$$\mathrm{NTU} = \frac{h_\mathrm{d} A_\mathrm{v} V}{G} = \int_{W_i}^{W_e} \frac{\mathrm{d}W}{W_\mathrm{sw} - W} \tag{9}$$

The cooling tower effectiveness (ε) is defined as the ratio of the actual energy transfer to the maximum possible energy transfer:

$$\varepsilon = \frac{h_\mathrm{e} - h_\mathrm{i}}{h_{\mathrm{sw}_\mathrm{i}} - h_\mathrm{i}} \tag{10}$$

Temperature ratio (TR) is defined as the ratio between actual range and ideal range and it is expressed as:

$$\mathrm{TR} = \frac{t_\mathrm{wi} - t_\mathrm{we}}{t_\mathrm{wi} - t_\mathrm{wbi}} \tag{11}$$

the exergy balance of an open system is:

$$\Sigma X_\mathrm{in} - X_\mathrm{D} = \Sigma X_\mathrm{out} \tag{12}$$

Total exergy of material stream is given by [11]:

$$X = X_\mathrm{PH} + X_\mathrm{KN} + X_\mathrm{PT} + X_\mathrm{CH} \tag{13}$$

By neglecting kinetic and potential energies, the total exergy is:

$$X = X_\mathrm{PH} + X_\mathrm{CH} \tag{14}$$

where, specific physical (thermomechanical) and chemical exergy is [12]:

$$x_\mathrm{PH} = x_\mathrm{tm} = x_\mathrm{mech} + x_\mathrm{thermal} = (h - h_\mathrm{o}) - T_\mathrm{o}(s - s_\mathrm{o}) \tag{15}$$

The specific chemical exergy defined in Wark [19] is shown as:

$$x_\mathrm{CH} = \sum_{k=1}^{n} \mathrm{x}_\mathrm{k} (\mu_\mathrm{ko} - \mu_\mathrm{koo}) \tag{16}$$

where x_k is the mole fraction of substance k in the mixture and μ is the chemical potential.

Specific exergy for psychometric process is:

$$x = (h - h_\mathrm{o}) - T_\mathrm{o}(s - s_\mathrm{o}) + \sum_{k=1}^{n} \mathrm{x}_\mathrm{k} (\mu_\mathrm{ko} - \mu_\mathrm{koo}) \tag{17}$$

On the basis of dry air and water vapor as an ideal gas, an alternative formula presented in Bejan [9]:

$$X_\mathrm{air} = G\left\{(C_\mathrm{pa} + W C_\mathrm{pv})\left[T - T_\mathrm{o} - T_\mathrm{o} \ln\frac{T}{T_\mathrm{o}}\right] + R_\mathrm{a} T_\mathrm{o}\left[(1 + 1.608W)\ln\frac{1 + 1.608W_\mathrm{oo}}{1 + 1.608W} + 1.608W \ln\frac{W}{W_\mathrm{oo}}\right]\right\} \tag{18}$$

By considering water as an incompressible fluid [19], on the basis of eq. (17) the exergy of water (X_W) represented as:

$$X_\mathrm{W} = L[(h_\mathrm{fw} - h_\mathrm{fo}) + v_\mathrm{ft}(P - P_\mathrm{st}) - T_\mathrm{o}(s_\mathrm{fw} - s_\mathrm{fo}) - R_\mathrm{v} T_\mathrm{o} \ln\varphi_\mathrm{o}] \tag{19}$$

The second term of eq. (19) is generally neglected when compared with $R_\mathrm{V} T_\mathrm{o} \ln\varphi_\mathrm{o}$:

$$X_\mathrm{W} = L[(h_\mathrm{fw} - h_\mathrm{fo}) - T_\mathrm{o}(s_\mathrm{fw} - s_\mathrm{fo}) - R_\mathrm{v} T_\mathrm{o} \ln\varphi_0 \tag{20}$$

Second-law efficiency is expressed as [9]:

$$\eta_{\text{II}} \quad \frac{\text{Total exergy out}}{\text{Total exrgy in}} \tag{21}$$

Using eq. (12) the second law efficiency is [14]:

$$\eta_{\text{II}} \quad 1 \quad \frac{\text{Energy destruction}}{\text{Exergy in}} \tag{22}$$

where exergy destruction (X_{D}) is:

$$X_{\text{D}} = (X_{\text{a in}} + X_{\text{w in}} + X_{\text{makeup}}) - (X_{\text{a out}} + X_{\text{w out}}) \tag{23}$$

$$\eta_{\text{II}} \quad 1 \quad \frac{X_{\text{D}}}{X_{\text{a in}} \quad X_{\text{w in}} \quad X_{\text{makeup}}} \tag{24}$$

Following constant values of air and water vapor are used: $R_{\text{a}} = 0.287$ kJ/kgK, $R_{\text{v}} = 0.461$ kJ/kgK, $C_{\text{pa}} = 1.003$ kJ/kgK, $C_{\text{pv}} = 1.872$ kJ/kgK. The dead state (ambient condition) conditions used for exergy analysis are $T_{\text{o}} = 25$ °C, $P_{\text{o}} = 101325$ Pa, and $W_{\text{oo}} = 0.009923$ kg$_{\text{w}}$/kg$_{\text{a}}$ ($\varphi = 50\%$). The results are plotted in figs. 2-10.

Results and discussion

For analysis purpose mass flow rate ratios are varied from 0.5 to 2.0 at an interval of 0.5 and air flow rate (G) is kept constant. The plots are generated for following set of input data: $t_{\text{db i}} = 35$ °C, $G = 270.46$ kg/s, $V = 645.81$ m^3, $h_{\text{d}}A_{\text{V}} = 1.2857$ kg/m^{3}s [19].

Rating analysis

This analysis shows the variation of outlet condition of water with inlet condition of air and water for the fixed tower volume. Figures 2 and 3 shows the variation of outlet water temperatures with respect to inlet air WBT and inlet water temperatures. Outlet water temperature ($t_{\text{w e}}$) increases with inlet air wet bulb temperature ($t_{\text{wb i}}$), inlet water temperature ($t_{\text{w i}}$), and also with increase in L/G ratio. Lowest $t_{\text{w e}}$ is achieved at lowest L/G ratio considered and the values of increasing rate for L/G ratio 1.0, 1.5, and 2.0 are 6.29, 4.85, and 3.44 °C for $t_{\text{wb i}}$ 1.34,

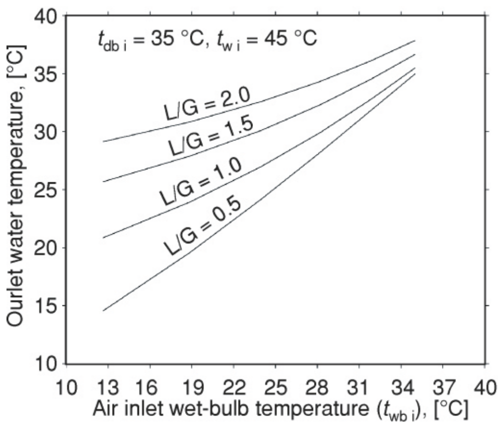

Figure 2. Variation of inlet air WBT with $t_{\text{w e}}$

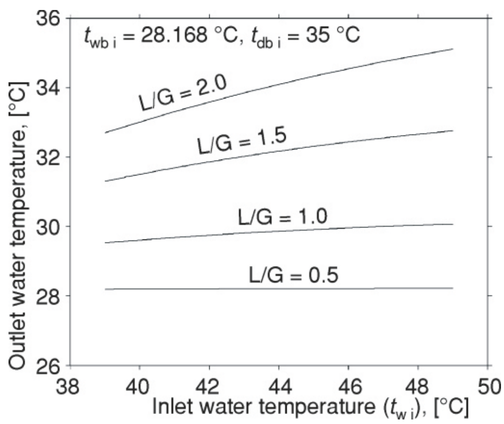

Figure 3. Variation of $t_{\text{w i}}$ with $t_{\text{w e}}$

1.77, and 1.4 °C for $t_{w\,i}$, respectively. This is due to increase in water flow rate for the same tower configuration, resulting in decreased cooling range and reduced heat transfer rates. Evaporation loss is decreases with increase in $t_{wb\,i}$ and increases with $t_{w\,i}$. For a particular L/G ratio, the change in $t_{w\,e}$ with respect to $t_{w\,i}$ is less when compared with changes with respect to inlet air wet bulb temperatures. Inlet air WBT has relatively more effect on outlet water temperature than inlet water temperature.

Figure 4 shows the effect of inlet air wet bulb temperature on water approach temperature. Difference between outlet water temperature and inlet air wet bulb temperature is termed as water approach temperature and it is high at lower $t_{wb\,i}$ when compared to higher $t_{wb\,i}$ at same L/G ratio. Water approach temperature increases with L/G ratio due to increase in heat load which leads to decrease in cooling range. For example, water approach temperature is 8.253 and 13.104 °C at L/G ratio 1 and 1.5 for the inlet air wet bulb temperature of 12.596 °C.

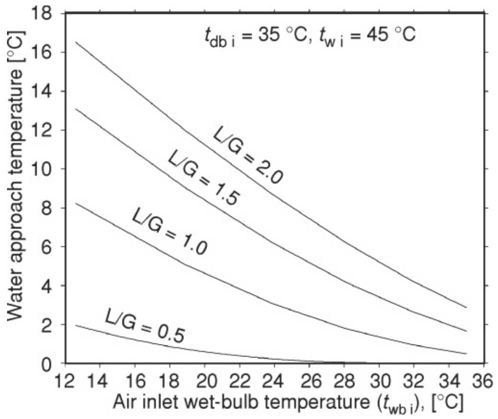

Figure 4. Variation of $t_{wb\,i}$ with water approach temperature

Energy analysis

Figures (5, 6) shows the effect on tower effectiveness as a function of inlet air WBT and inlet water temperature for different mass flow rate ratios considered. Effectiveness decreases with increase in $t_{wb\,i}$ and $t_{w\,i}$. As L/G ratio increases, ε increases, but increasing rate of ε is decreases with increase in L/G ratio. For the $t_{wb\,i}$ and $t_{w\,i}$ values considered, the ε is 0.5869 for 12.596 °C, 0.4961 for 35 °C and 0.6029 for 39 °C, 0.4813 for 49 °C at L/G ratio 1.0, respectively. From fig. 6 it is inferred that, changes in inlet water temperature has relatively more effect on effectiveness of the cooling tower than $t_{wb\,i}$.

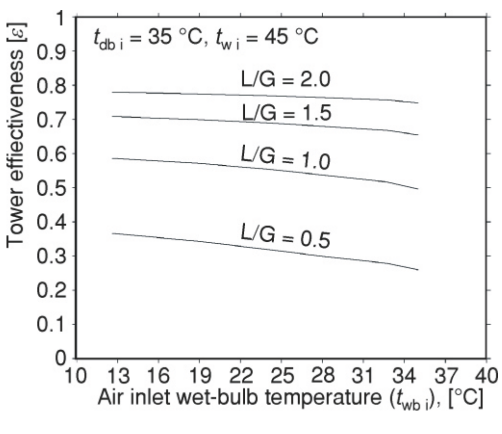

Figure 5. Variation of $t_{wb\,i}$ with tower effectiveness

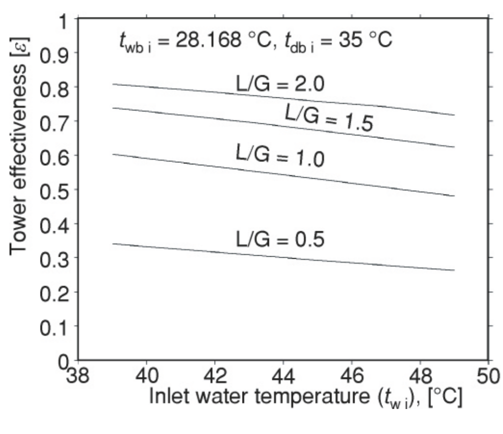

Figure 6. Variation of $t_{w\,i}$ with tower effectiveness

Figure 7 shows the effect on temperature ratio as a function of inlet air wet bulb temperature. Temperature ratio decreases with increase in L/G ratio. This is due to increase in heat load which leads to lesser cooling range. At lower L/G ratio actual cooling range approaches ideal range. For example, temperature ratio is 0.7453 and 0.5956 for L/G ratio 1 and 1.5 at $t_{\mathrm{wb\,i}}$ 12.596 °C.

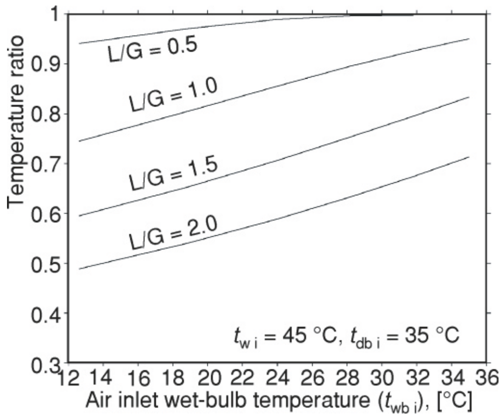

Figure 7. Variation of $t_{\mathrm{wb\,i}}$ with temperature ratio

Exergy analysis

Figure 8 shows the effect on second law efficiency (η_{II}) as a function of inlet air wet bulb temperature for different mass flow rate ratios. The second law efficiency η_{II} increases with increase in $t_{\mathrm{wb\,i}}$. Increase in η_{II} shows the decreasing rate of exergy destruction (X_{D}). Exergy destroyed decreases with increasing $t_{\mathrm{wb\,i}}$ due to increasing $t_{\mathrm{wb\,i}}$ towards $t_{\mathrm{db\,i}}$. As $t_{\mathrm{wb\,i}}$ increases, exergy of makeup water decreases due to decrease in evaporation loss. Exergy of water at inlet is constant and at outlet it increases due to increase in $t_{\mathrm{wb\,i}}$ and exergy of inlet air is increases and at outlet also increases continuously due to higher outlet air DBT ($t_{\mathrm{db\,e}}$) and humidity ratio (W) that are achieved. These factors lead to increase in η_{II} and it can be observed from decreasing value of water approach temperature shown in fig. 7. For the $t_{\mathrm{wb\,i}}$ values considered, the η_{II} is 93.46% for 12.596 °C and 98.419% for 35 °C and the corresponding exergy destruction (entropy generation or irreversibility) is 1800.2 kW and 436.76 kW at L/G ratio 1.0.

Figure 9 shows the effect on second law efficiency as a function of inlet water temperature for different mass flow rate ratios. It is noticed that X_{D} increases and η_{II} decreases for the increase in $t_{\mathrm{w\,i}}$ and exergy of air at inlet is constant and at outlet it increases due to constant increase in inlet water temperature. Exergy of makeup water increases with $t_{\mathrm{w\,i}}$ due to increase in evaporation loss since difference between inlet air wet bulb temperature and inlet water temperature increases. Decrease in cooling range leads to increase in exergy destruction. For the $t_{\mathrm{w\,i}}$ values considered, the η_{II} is found to be 98.09% for 39 °C and 96.412% for 49 °C at L/G ratio 1.0.

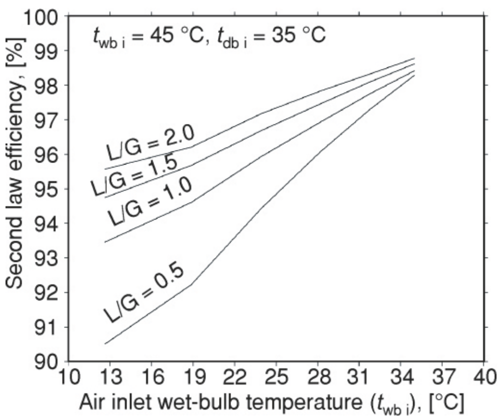

Figure 8. Variation of second law efficiency with $t_{\mathrm{wb\,i}}$

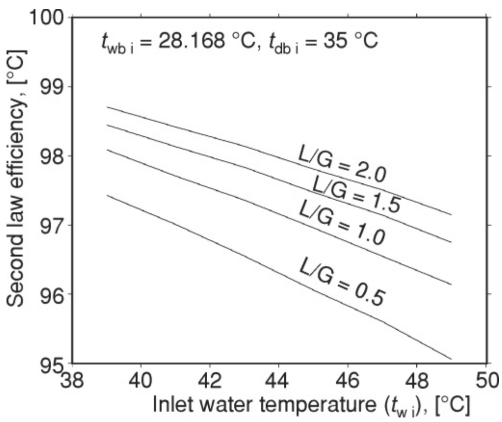

Figure 9. Variation of second law efficiency with $t_{\mathrm{w\,i}}$

Exergy of air is divided in to exergy of air via convection and evaporation. Convection air exergy is function of dry bulb temperature and evaporation is function of humidity ratio of wet air moving from bottom to top of the tower. Total exergy of air is sum of convection air exergy and evaporation air exergy. Variation of air exergy with size of the tower for the mass flow rate ratios considered is shown in figs. 10 and 11.

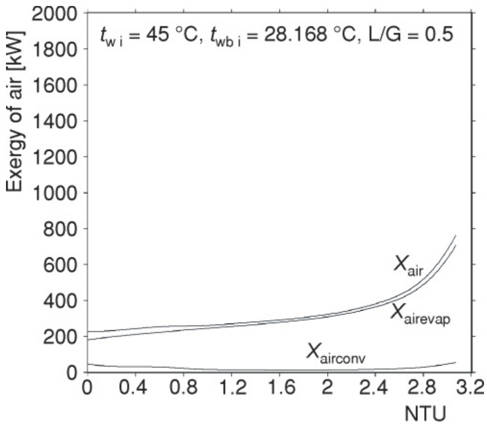

Figure 10. Variation of air exergy with size of the tower for L/G = 0.5

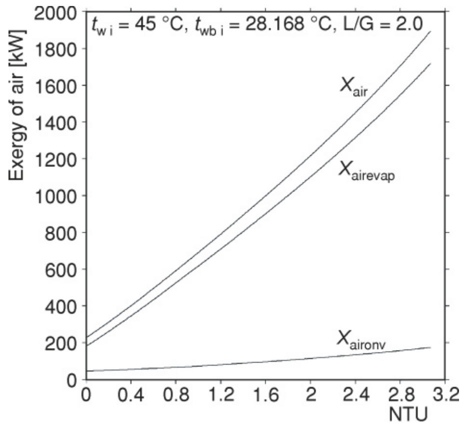

Figure 11. Variation of air exergy with size of the tower for L/G = 2.0

Exergy of air moving from bottom to top of the tower is described by eq. (18) where the first term represents the convective air exergy ($X_{airconv}$) and second term represents the evaporative exergy ($X_{airevap}$). Along the size of the tower convective air exergy decreases up to some height from bottom and then increases while reaching the top of the tower. Decrease in $X_{airconv}$ shows the negative convection.

As expected the variation of $X_{airconv}$ is same as variation of dry bulb temperature. Here evaporative air exergy always increases with the size of the tower which can be understood from the fact that humidity ratio is increasing from bottom to top of the tower. It is also clearly shown in above figures the process is always dominated by air via evaporation.

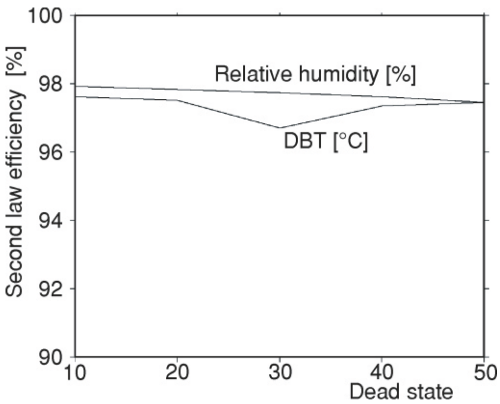

Figure 12. Effect of dead state on second law efficiency

Dead state variation

Figure 12 shows the effect on second law efficiency as a function of ambient conditions (dead state). To generate this plot, dead state DBT is varied from 10 to 50 °C at an interval of 10 °C and dead state relative humidity is varied from 10 to 50% at an interval of 10%. It is noticed that the change in second law efficiency with respect to variation in dead state is not significant in both cases [7]. Maximum difference of 0.91% and 0.47% in second law efficiencies for varying dead state DBT and relative humidity.

Conclusions

At lower L/G ratio, actual cooling range approaches ideal cooling range of counter flow wet cooling tower. This paper establishes, at lower inlet air WBT, the outlet water temperature decreases which leads to higher water approach temperature and exergy destruction there by decreases the second law efficiency. At higher inlet air wet bulb temperature, the outlet water temperature increases which leads to decreases both the water approach temperature and exergy destruction which leads to higher second law efficiency. Air exergy by evaporation mode always controls the exergy of air. For a 22.4 °C rise in $t_{wb\,i}$, the ε decreases by 9.08%, TR increases by 0.2057 and η_{II} by 4.959%. For 10 °C rise in $t_{w\,i}$, the ε is decreased by 12.16% and η_{II} by 1.948% at L/G = 1.0. Furthermore, it is noticed that, dead state has insignificant effect on second law efficiency.

Nomenclature

A_V – surface area of water droplets per unit volume of tower, [m^2m^{-3}]
C_p – specific heat capacity at constant pressure, [$kJkg^{-1}K^{-1}$]
G – mass flow rate of dry air, [kgs^{-1}]
h – specific enthalpy, [$kJkg^{-1}$]
h_c – convective heat transfer coefficient of air, [$kWm^{-2}K^{-1}$]
h_d – convective mass transfer coefficient, [$kgWm^{-2}s^{-1}$]
h_f – specific enthalpy of saturated liquid water, [$kJkg_w^{-1}$]
$h_{f\,w}$ – specific enthalpy of water at t_w, [$kJkg_w^{-1}$]
$h_{fg\,w}$ – change of phase enthalpy ($h_{fg} = h_{gw} - h_f$), [$kJkg_w^{-1}$]
h_g – specific enthalpy of saturated water vapor, [$kJkg_w^{-1}$]
$h_{g\,w}$ – specific enthalpy of water vapor, [$kJkg_g^{-1}$]
h_g^0 – specific enthalpy of saturated water vapor evaluated at 0 °C, [$kJkg_w^{-1}$]
$h_{s\,w}$ – enthalpy of saturated moist air evaluated at t_w, [$kJkg_a^{-1}$]
L – mass flow rate of water, [kg_ws^{-1}]
Le – Lewis number (= h_c/h_dC_{pa}), eq. 4, [–]
P – pressure, [Pa]
$P_{sat\,d}$ – presure at saturation temperature, [Pa]
R – gas constant, [$kJkg^{-1}K^{-1}$]
T – dry bulb temperature, [K]
t – dry bulb temperature of moist air [°C]
t_w – water temperature, [°C]
s – specific entropy, [$kJkg^{-1}K^{-1}$]
V – volume of tower, [m^3]
v – specific volume, [m^3kg^{-1}]
W – humidity ratio of moist air, [$kg_wkg_a^{-1}$]
X – total exergy, [kW]
x – specific exergy, [$kWkg^{-1}$]
x – mole fraction of the substance, [kmolkg–1]

Greek symbols

ε – effectiveness, [–]
μ – chemical potential, [$kJkmol^{-1}$]
– relative humidity, dimensionless
η_{II} – second law efficiency

Abbreviations

DBT – dry bulb temperature, [°C]
NTU – number of transfer units
TR – temperature ratio
WBT – wet bulb temperature [°C]

Subscripts

a – moist air
CH – chemical
db – dry bulb
D – destruction
e – outlet
g w – vapor at water temperature
i – inlet
KN – kinetic
o – restricted dead state
oo – dead state
PH – physical
PT – potential
s w – saturated moist air at water temperature
w – water
wb – wet bulb

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