ABSTRACT
In this paper, performance evaluation of wet cooling tower is done. To achieve this aim, first, thermal behavior of counter-flow wet cooling tower is studied through a simulation model. The influence of the environmental conditions on the thermal efficiency of the cooling tower is investigated. The cooling tower performance is simulated in terms of varying air and water temperatures, and of the ambient conditions. This model allows the use of a variety of packing materials. Second, the exergetic analysis is applied to study the cooling tower potential of performance improvement. The model is validated against the experimental data.

INTRODUCTION
Cooling towers are widely used in energy systems and industrial processes to dissipate waste heat from hot process streams into the environment. Heat ejection from the wet cooling tower occurs as convetional transfer between water droplets and the surrounding air, and also as the evaporation of a small portion of the water into the moving air. Therefore, the process involves both heat and mass transfer.

Several works have been done to investigate the performance of cooling towers over the last century. Lemouari et al. investigated the thermal performance of cooling tower. Khan et al. performed a risk based approach to analyze the performance of cooling tower through a fouling model. Soylemez presented a thermo-hydraulic performance optimization method of mechanical draft cooling tower. Also various references contain examples that illustrate application of the second law of thermodynamics to cooling systems. Qureshi and Zubair studied the second-law-based evaluation of cooling towers and evaporative heat exchangers under varying operating conditions. In their paper, the improvement of cooling tower performance achieved by changing the water and ambient air temperature. It is important to note that the inlet water temperature of a cooling tower and the environment temperature are the fixed parameters when dealing with an existing cooling water system. This is because, the water that enters the cooling tower is provided by the heat-exchanger network. Furthermore, the air temperature varies with changes in the environmental conditions. Muangnoi et al. presented exergy analysis of cooling towers. They demonstrated the exergy change along the cooling tower. However, little attention has been placed to the application of exergetic analysis to performance improvement of cooling tower.

In this study, first, the thermal performance of cooling tower under different operational and environmental conditions has been studied. Second, the exergy analysis has been applied to evaluate the performance of cooling tower on the basis of the second law of thermodynamics. To achieve this objective a mathematical model of counter flow wet cooling tower has been introduced. This model allows prediction of tower performance by using the heat and mass transfer between water and air to drive the solution to steady state conditions. In this model, the second law is used to explore the exergy distribution of water and air in cooling tower. Moreover, it involves the calculation of system performance in form of the second law efficiency.

MATERIALS AND METHODS
Mathematical model of cooling tower
The total enthalpy transfer at the air-water interface consists of an enthalpy transfer associated with the mass transfer due to the difference in vapour concentration, and a heat transfer due to the difference in temperature (Kim and Smith, 2001). The heat and mass transfer between the air and water within the cooling tower's packing material is illustrated in Figure 1. The following mathematical model entails the following assumptions:

1. Heat and mass transfer through the tower wall to the environment is negligible.
2. The flow rates of dry air and water are constant.
3. Temperatures of water and air are uniform at any cross section.
4. Temperature has no influence on the transfer coefficients.
5. Water loss by drift is negligible.
6. Interface areas for heat and mass transfer are equal.

Figure 1, Control volume of counter flow tower.
The water energy balance in terms of heat and mass transfer coefficients are given by Eq. 1.

\[ m_w \frac{dh_{fw}}{dH} + m_a \frac{dhw_{fw}}{dH} = h_c A_v (T_w - T) dV + h_d A_v h_{fg,w} (w_{sw} - w) dV \]  \hspace{1cm} (1)

At the steady state condition, energy balance between air and water considering the evaporation yields Eq. 2.

\[ m_a \frac{dh}{dh_{fw}} = (m_{w,\text{out}} - m_a (w_{in} - w)) dh_{fw} + m_a \frac{dhw_{fw}}{dH} \]  \hspace{1cm} (2)

The convective mass transfer is given by Eq. 3 (Khan et al., 2003).

\[ m_a dw = h_d A_v dV (w_{sw} - w) \]  \hspace{1cm} (3)

The energy removed by water is given by Eq. 4.

\[ dQ_{\text{ref}} = m_w C_{pw} dT_w \]  \hspace{1cm} (4)

By substituting Lewis factor as \( Le_f = \frac{h_c}{h_d C_{pa}} \) in Eq. 1, it simplified to Eq. 5.

\[ m_w \frac{dh_{fw}}{dH} + m_a \frac{dhw_{fw}}{dH} = h_d A_v dV [Le_f C_{pa} (T_w - T) + (w_{sw} - w)h_{fg,w}] \]  \hspace{1cm} (5)

By combining the Eqs. (1)-(4), and substituting \( ka = h_d A_v \), it yielded Eq. 6 (Khan and Zubair, 2001).

\[ \frac{dh}{dH} = \frac{KaA}{m_a} [Le_f C_{pa} (T_w - T) + h_{gw} (w_{sw} - w)] \]  \hspace{1cm} (6)

Where \( KaA \) is the cooling tower performance characteristic (Kröger and Kloppers, 2005). The cooling tower characteristic is expressed as Eq. 7 (Kröger, 2004).

\[ \frac{KaV}{L} = \frac{\tau_{w,\text{in}} C_{pw} dT_w}{\tau_{w,\text{out}} (h_{fw} - h_{fg,w})} = \frac{L_f h_d A_v}{m_w} \]  \hspace{1cm} (7)

By substituting tower characteristics in Eq. 3, the humidity change along the cooling tower is as Eq. 8.

\[ \frac{dw}{dH} = \frac{KaA}{m_a} (w_{sw} - w) \]  \hspace{1cm} (8)

The energy gained by air in the incremental volume has been written as Eq. 9 (Khan and Zubair, 2004).

\[ dQ_a = m_w dh_{fw} + h_{fw} m_a dw \]  \hspace{1cm} (9)

The cooling water temperature along the tower is expressed as Eq. 10 (Muangnoi et al., 2007).

\[ dT_w = \frac{m_a}{m_w C_{pw}} (dh - h_{fw} dw) \]  \hspace{1cm} (10)

To account the effect of changing fill type in the cooling tower performance characteristic:
The $a_d$ and $b_d$ are the packing transfer coefficients which are different for each type of packing (Koger, 2004). Therefore the water outlet condition can be calculated for different packing.

The above mentioned wet cooling tower mathematical modeling allows the use of a variety of packing materials in the cooling tower toward optimizing cooling tower performance. In this study, the computations have been done through a flow chart and the related coding was developed to achieve the result of calculations. The algorithm of the same is presented in Figure 2.

**Exergetic analysis**

The total exergy of system is defined in terms of thermo-mechanical and chemical exergy (Eq.12). The thermo-mechanical exergy is written as Eq. 13 (Qureshi and Zubair, 2007).

$$\psi = \psi_{tm} + \psi_{ch}$$

$$\psi_{tm} = (h - h_0) - T_o (S - S_0)$$

The specific chemical exergy is given as Eq. 14 (Wark, 1995):

$$\psi_{ch} = \sum_{k=1}^{n} x_k (\mu_{k,0} - \mu_{k,env})$$
In cooling tower, air and water incorporates the cooling performance. Therefore, the effects of both air and water have been studied in the exergy analysis. The total water exergy is expresses as Eq. 15.

\[
X_w = m_w \left[ (h_{f,w} - h_{f,0}) - T_0 (S_{f,w} - S_{f,0}) - R_v T_0 \ln \phi \right]
\]  

(15)

The total exergy of air is given through Eq. 16.

\[
X_a = x_a \left[ \Delta h_a - \Delta h_{a,0} - T_0 (\Delta S_a - \Delta S_{a,0}) + \Delta \mu_a - \Delta \mu_{a,0} \right] \\
+ \ x_v \left[ \Delta h_v - \Delta h_{v,0} - T_0 (\Delta S_v - \Delta S_{v,0}) + \Delta \mu_v - \Delta \mu_{v,0} \right]
\]  

(16)

In Eq. 16, the enthalpy, entropy and the chemical potential are presented in mole basis units which are achieved considering the dead state. Below definitions have been used for the above terms, Eq. 17-19. The over-bar (\(\bar{\cdot}\)) represents the mole basis (Muangnoi et al., 2007).

\[
\Delta \bar{h}_i = \bar{C}_p \bar{v} (T - T_0)
\]  

(17)

\[
\Delta \bar{S}_i = \bar{C}_p \ln \frac{T}{T_0} - R_v \ln \frac{P}{P_0}
\]  

(18)

\[
\Delta \bar{\mu}_i = \bar{R} T_0 \ln \frac{x_i}{x_{i,0}}
\]  

(19)

The total air exergy includes terms of exergy related to the convective heat transfer and the evaporative heat transfer. The air exergy through convective heat transfer is written as Eq. 20.

\[
X_{a,c} = (x_a \bar{C}_{pa} + x_v \bar{C}_{pv}) \left( T - T_0 - T_0 \ln \frac{T}{T_0} \right) + \bar{R} T_0 \ln \frac{P}{P_0}
\]  

(20)

The specific heat of dry air and water vapor, \(C_{pa}\) and \(C_{pv}\), in Eq. 20 can be calculated through Eq. 21 and Eq. 22 respectively (Kroger, 2004).

\[
C_{pa} = 1.045356 \times 10^3 - 3.161783 \times 10^{-1} T + 7.083814 \times 10^{-4} T^2 \\
- 2.705209 \times 10^{-7} T^3
\]  

(21)

\[
C_{pv} = 1.3605 \times 10^3 + 2.31334 T - 2.46784 \times 10^{-10} T^5 + 5.91332 \times 10^{-13} T^6
\]  

(22)

The air exergy by the evaporative heat transfer is given as Eq. 23 (Muangnoi et al., 2007).

\[
X_{a,e} = \bar{R} T_0 \left( x_a \ln \frac{x_a}{x_{a,0}} + x_v \ln \frac{x_v}{x_{v,0}} \right)
\]  

(23)

Exergy consumption is accompanied by entropy generation (Krakow, 1994). Therefore, the generated entropy must be discarded continuously from water. The generated entropy is proportional to exergy loss. To achieve the exergy destruction, \(X_D\), the loss potential of air to be recovered by water can be constructed through the exergy balance of the elementary control volume (Eq. 24) (Cheng Qin et al., 2002).
\[ X_D = \sum_{in} X - \sum_{out} X = (X_{w,in} + X_{a,out}) - (X_{w,out} + X_{a,in}) \]  \( (24) \)

The second law efficiency, which is a measure of irreversible losses in a given process, is defined as Eq. 25 (Bejan, 1997).

\[ \eta = \frac{\sum_{in} X}{\sum_{out} X} = 1 - \frac{X_D}{\sum X} \]  \( (25) \)

RESULTS AND DISCUSSION
Verification of the proposed simulation model
We tested our method by applying the experimental data of Simpson and Sherwood (1946) to the cooling tower model.

The experimental data are related to a counter-flow wet cooling tower. The most comparable results are those of the exit water temperatures and the exit wet-bulb temperatures. The simulation results are presented in Table 1.

<table>
<thead>
<tr>
<th>Experimental Data</th>
<th>Case 1</th>
<th>Case 2</th>
<th>Case 3</th>
<th>Case 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water Inlet Temperature (K)</td>
<td>311.93</td>
<td>311.83</td>
<td>307.65</td>
<td>301.87</td>
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<tr>
<td>Water Outlet Temperature (K)</td>
<td>302.48</td>
<td>302.48</td>
<td>299.37</td>
<td>297.37</td>
</tr>
<tr>
<td>Air Inlet Dry Bulb Temperature (K)</td>
<td>308.15</td>
<td>308.15</td>
<td>303.65</td>
<td>302.15</td>
</tr>
<tr>
<td>Air Inlet Wet Bulb Temperature (K)</td>
<td>299.82</td>
<td>299.82</td>
<td>294.26</td>
<td>294.26</td>
</tr>
<tr>
<td>Air Outlet Dry Bulb Temperature (K)</td>
<td>306.42</td>
<td>306.42</td>
<td>303.42</td>
<td>299.82</td>
</tr>
<tr>
<td>Water Flow Rate (kg/s)</td>
<td>1.008</td>
<td>1.008</td>
<td>1.259</td>
<td>1.259</td>
</tr>
<tr>
<td>Air Flow Rate (kg/s)</td>
<td>1.250</td>
<td>1.250</td>
<td>1.187</td>
<td>1.187</td>
</tr>
<tr>
<td>Tower Area (m²)</td>
<td>1.057</td>
<td>1.057</td>
<td>1.068</td>
<td>1.043</td>
</tr>
</tbody>
</table>

Model Output Result
| Water Outlet Temperature           | 302.11 | 302.98 | 299.02 | 298.08 |
| Air Outlet Dry bulb Temperature (K) | 306.23 | 306.86 | 303.05 | 300.12 |
| Tower Area (m²)                    | 1.074  | 1.036  | 1.068  | 1.043  |

Result Error
| Water Outlet Temperature Error (%) | -0.12  | 0.16   | -0.11  | 0.23   |
| Air Outlet Dry Bulb Temperature (%) | -0.06  | 0.14   | -0.12  | 0.01   |
| Tower Area Error (%)              | 0.01   | -0.02  | 0.01   | -0.01  |

These results suggest that the proposed model is accurate based on the limited amount of available experimental data. Therefore, the model can be used to predict the properties of the exit water and air from the tower for a given design and operating conditions.

Performance simulation of wet cooling towers by the proposed model
It is usually important to supply cooling water at a specific temperature. However, the performance of a cooling tower will vary with changes in environmental conditions. This will affect the cooling water outlet temperature. Investigating the thermal behavior of the cooling tower at different environmental conditions enables the prediction of a tower’s performance at different atmospheric conditions. Figure 3 shows the effect of wet bulb temperature on water outlet temperature and evaporation loss for different liquid to gas ratios. The plots are drawn using the following set of input data: \( P_{atm} = 101325 \) Pa; \( T_{w,in} = 41 \) °C; \( m_g = 32.44 \) kg/s; \( H = 2.51 \) m.
As shown in Figure 3, the water outlet temperature increases when the environment wet bulb temperature is increased. The outlet conditions, flow rate, and temperature of the water are affected by evaporation. Figure 3 also demonstrates that reducing the wet bulb temperature results in increasing evaporation loss. When the wet bulb temperature is 16 °C and the liquid to gas ratio of tower is 1.5, the tower can supply cooling water at a temperature of 32.8 °C. However, with an increase of 2.3 °C in the environment wet bulb temperature (18.3 °C), the temperature of the cooling water from the tower increases to 34.25 °C. This affects the performance of the cooling system. Therefore, to provide cooling water at a temperature of 32.8 °C under the new environmental conditions, the liquid to gas ratio needs to be decreased to 1.1.

The cooling tower approach is defined as the difference between the water outlet temperature and the wet bulb temperature (Kim et al., 2001; Panjeshahi and Ataei, 2008). Figure 4 shows the isothermal cooling line of the cooling system outlet temperature. The graphs are drawn for different approach values of 5 °C, 8 °C, and 11 °C. It is shown that if the temperature of the cooling water outlet remains constant, the water inlet temperature needs to be reduced when the water flow rate increases. Moreover, decreasing water flow rate and increasing water inlet temperature simultaneously results in reducing the water outlet temperature.

The cooling tower's heat ejection versus water inlet flow rate at different inlet temperatures is shown in Figure 5. It demonstrates that when the water flow rate is decreased by 4 kg/s, the heat removal accomplished by the tower increases by 74 kW for a water inlet temperature of 45 °C. The rate of heat ejection continues to increase at higher water inlet temperatures. In other words, when the inlet cooling water has a high temperature and low flow rate, the tower ejects more heat from the water (Panjeshahi and Ataei, 2008).
Figure 5. Heat removal versus inlet water flow rate at different inlet temperatures.

Figure 6 shows the variation of evaporation rate versus heat removal. The water flow rate is set at 16.58 kg/s. It can be seen from the figure that the evaporation rate increases as heat removal increases, and that a constant heat ejection value does not necessarily ensure a fixed evaporation rate. The amount of evaporation depends on the air flow rate, the humidity of the inlet air, and the humidity of the cooling tower outlet air. The exit air humidity is interconnected with the water temperature and the transfer area of the packing material.

Figure 6. Variation of evaporation rate with heat removal at different liquid to gas ratios.

Figure 7 shows the variation of the tower characteristic $M_e$, with the inlet water temperature for liquid to gas ratios of 0.5, 1.1, and 1.5. The figure demonstrates that this tower characteristic decreases with an increase of L/G. In other words, the tower $M_e$ is higher for the lower L/G values, corresponding to the lower water flow rate, which results the best cooling.

Figure 7. Variation of tower characteristic $M_e$ with water inlet temperature.
Figure 8 shows cooling tower performance in terms of effectiveness. A high degree of tower effectiveness corresponds to better cooling performance and higher heat removal. It can be seen in Figure 8 that when the inlet cooling water has a high temperature and low flow rate, the effectiveness of the cooling tower increases. This confirms the experimental results of Bedekar et al. (1998).

![Figure 8. Effects of inlet water flow rate and temperature on tower effectiveness.](image)

Performance evaluation of wet Cooling towers by exergy analysis

The proposed simulation model allows different packing material to be chosen in counter-flow mechanical draft wet cooling towers. Three packing types have been selected to carry out results of this analysis. To investigate the exergy analysis of cooling tower, the experimental data presented in case 4 from Table 1 are used. Also following data of air-vapor and ambient conditions have been used in the analysis:

\[
T_{db} = 298.15 \text{ K}, \quad T_{wb} = 294.24 \text{ K}, \quad R_a = 0.287 \frac{kJ}{kgK}, \quad R_v = 0.461 \frac{kJ}{kgK},
\]

\[
T_0 = 298.16 \text{ K}, \quad P_o = 101.325 \text{ kPa}, \quad \phi_o = 50\%.
\]

Figure 9 shows the temperature profile of moving water and air along the cooling tower. As shown in Figure 9, water cools as it flows downward and its temperature decreases. On the other side, air dry bulb temperature initially decreases as it enters the tower and it increases after tower height of 0.61 m (the intersection between water and air profile). This intersection point of dry bulb temperature and water profile indicates no temperature difference; hence, no convective heat transfer of air to water. It is noted from the Figure 9, that wet bulb temperature increases continuously from bottom to top of the tower and is always less than water temperature. As a result, heat flows from water into air. Therefore, heat transfer mode in cooling tower has been dominated by evaporation. This is clearly shown in Figure 10.

![Figure 9. Water and air temperature profile along the cooling tower.](image)

The exergy of air expresses the available energy of air to utilize that supplied by water. Figure 10, shows the exergy of air via convective and evaporative heat transfer along the cooling tower (Eq. 20-23). As shown the exergy corresponded to convective transfer decreases from bottom to the height of 0.61 m. This point indicated
minimum air exergy through convective heat transfer. After this point, the convective exergy contained in air is able to let thermal energy flow into it and as shown in Figure 9, dry bulb temperature increase. The results reveal the exergy of air via evaporative heat transfer increases along the tower.

![Figure 10, Air exergy along the cooling tower.](image)

Table 2 demonstrates the exergy of water and air via convective and evaporative heat transfer through height of the cooling tower.

<table>
<thead>
<tr>
<th>Tower Height (m)</th>
<th>0.1</th>
<th>0.3</th>
<th>0.5</th>
<th>0.7</th>
<th>0.9</th>
<th>1.1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Exergy of Water (kW)</td>
<td>119.1</td>
<td>119.3</td>
<td>115.5</td>
<td>119.7</td>
<td>119.9</td>
<td>120.1</td>
</tr>
<tr>
<td>Exergy of Air via Convective Transfer (kW)</td>
<td>$22 \times 10^{-3}$</td>
<td>$12 \times 10^{-3}$</td>
<td>$7 \times 10^{-3}$</td>
<td>$6 \times 10^{-3}$</td>
<td>$7 \times 10^{-3}$</td>
<td>$12 \times 10^{-3}$</td>
</tr>
<tr>
<td>Exergy of Air via Evaporative Transfer (kW)</td>
<td>$7.1 \times 10^{-2}$</td>
<td>$17 \times 10^{-2}$</td>
<td>$30 \times 10^{-2}$</td>
<td>$44 \times 10^{-2}$</td>
<td>$60 \times 10^{-2}$</td>
<td>$75 \times 10^{-2}$</td>
</tr>
</tbody>
</table>

As shown in Figure 10 and Table 2, the amount of exergy supplied by water is higher than that absorbed by air. Therefore, exergy contained in water is able to disperse its thermal energy into the environment. Also, it is indicated that air exergy is capable for thermal energy flow into it. This is because the entropy is generated by the system.

Figure 11 has illustrated the exergy destruction along the cooling tower for three types of packing. It is indicated the distribution of exergy loss is high at bottom and gradually low at the top of tower. Hence, minimum exergy destruction is accomplished at the top for each packing type. Moreover, it shows that the hexagonal corrugated packing type is resulted in less exergy destruction. This is due to large transfer area of moving water and air.

![Figure 11, Exergy loss along the cooling tower for the different packing types.](image)

The exergy loss is computed for the different inlet water temperature in three types of packing at constant air exergy. Figure 12 shows the variation of exergy loss versus inlet water temperature. The results of the model
are indicated that the exergy of water is increased due to the increase in inlet water temperature. However, the exergy of outlet water decreases. This is because of reducing exit water temperature as the result of increasing inlet water temperature. Therefore, the exergy destruction increases by increasing the inlet water temperature. As shown in Figure 12, using sinusoidal corrugated fill is resulted in higher exergetic loss in comparison to the hexagonal packing that has larger surface area. In other words, changing packing type from sinusoidal to hexagonal fill is accomplished 26% exergy loss reduction under the existing condition of case 4.

![Exergy loss versus inlet water temperature at the different packing types.](image)

Figure 12, Exergy loss versus inlet water temperature at the different packing types.

Figure 13 illustrates the second law efficiency change versus the variation of cooling tower inlet water temperature. It is noted from the results that second law efficiency decreased as the exergy loss increased by increasing of the inlet water temperature. Furthermore, it is demonstrated that hexagonal fill made higher efficiency relative to triangular and sinusoidal corrugated packing types. The efficiency achieved through using hexagonal fill is 0.2% higher than the sinusoidal fill at the presented operating condition in case 4. The exergetic analysis results of cooling tower are revealed that large heat and mass transfer surface increases the second law efficiency. Moreover the bottom of cooling tower provided greater opportunity for improvement due to higher exergetic destruction. Therefore, the packing with large transfer area, hexagonal type, at the bottom of tower is resulted in improvement of the performance by decreasing exergy loss and increasing efficiency. Evaluation of cooling tower through exergy loss analysis is accomplished to optimal performance and consequently the environmental benefits through selection of optimal operating conditions and appropriate packing type.

![Second law efficiency versus inlet water temperature at the different packing types.](image)

Figure 13, Second law efficiency versus inlet water temperature at the different packing types.

**CONCLUSION**

Evaluation of cooling tower performance is explored through exergetic analysis. To achieve this aim, a mathematical model of counter flow wet cooling tower is developed to predict water and air properties. The proposed simulation model is validated against experimental data. It is noted that the errors between the
The predicted and experimental values are achieved within 0.14%. The influence of operational and atmospheric conditions on thermal behaviour of cooling tower is studied. Moreover, the effect of tower characteristics on the basis of packing type is studied. Also, this model allowed the exergetic analysis of water and air along cooling tower through the fundamental balance law. The results of the cooling tower modeling illustrated that the amount of exergy supplied by water is larger than that absorbed by air. This is because the entropy is generated by the system. To depict the utilisable exergy between water and air, the exergy of each working fluid along the tower has been presented. The results revealed that the water exergy decreases continuously from the top to bottom. On the other hand, the air exergy has been expressed in terms of convective and evaporative heat transfer.

Furthermore, the distribution of the exergy destruction has been used as a guideline to find the optimal potential for improving the cooling tower performance and reducing the environmental impacts of the tower. The results revealed that the potential of improvement is higher at the bottom of tower. Therefore, using packing with large heat and mass transfer area (corrugated hexagonal packing type) is resulted in less exergetic loss in comparison to the other packing types. In other words, by changing the packing type from sinusoidal to hexagonal fill, 26% exergy loss reduction is achieved under the fixed operational conditions. Therefore, the efficiency achieved through using hexagonal fill is 0.2% higher than the sinusoidal fill and more environmental benefits as the result of better performance of cooling tower.

**NOMENCLATURE**

<table>
<thead>
<tr>
<th>Symbol</th>
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<tr>
<td>$a_c, b_d$</td>
<td>Transfer coefficient constants</td>
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<tr>
<td>A</td>
<td>tower cross-sectional area, m$^2$</td>
</tr>
<tr>
<td>$A_{LV}$</td>
<td>air-water interfacial area per unit volume of tower, m$^2$ m$^3$</td>
</tr>
<tr>
<td>$C_{pa}$</td>
<td>specific heat of dry air at constant pressure, kJ/kg K</td>
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<tr>
<td>$C_{pv}$</td>
<td>specific heat of water vapor at constant pressure, kJ/kg K</td>
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<td>heat transfer coefficient of air, kW/m$^2$K</td>
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<td>$L_f$</td>
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**Greek letters**

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<td>$\mu$</td>
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<td>specific exergy, kJ/kg</td>
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<td>relative humidity</td>
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**Subscripts**

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REFERENCES