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*Research Article*

# Thermal and Exergy Analysis of Counter Flow Induced Draught Cooling Tower

**Abdul Hadi N. Khalifa**[†*]

[†]Engineering Technical College, Middle Technical University. Baghdad-Iraq

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***Abstract***

*Merkel theory was derivative in this work for a counter flow induced draught cooling tower with some simplifications. The governing equation was solved by an iterative method. The tower was divided into 100 horizontal elements, the temperature difference in each element was 0.1 K. Mass, energy and exergy balances were evaluated for each element using Engineering Equation Solver (EES) software. For such tower, it was found that the ratio of mass flow rate of water to that for air (L/G) is in the range of 1.25 to 1.5. Since the exergy of air is consumed to destroy the exergy of water, then, more exergy destruction gives higher exergy efficiency for cooling tower. As the moisture content of the air increases the air chemical exergy increases, on the another hand, as air temperature approaches to water temperature air thermal exergy tend to reduce. And finally, Merkel assumption, that state that saturated air leaving cooling tower, gives curvature path for saturation process instead of a straight line.*

***Keywords:*** *Cooling Tower, Exergy, Heat and Mass Transfer, Merkel theory , exergy destruction, Thermal Performance.*

## 1. Introduction

Energy and mass transfer through cooling tower play significant roles in cooling water and concerned many researchers, the first one who developed the theory of cooling tower was Merkel (Merkel, 1925). Merkel assumed that the energy and mass in cooling tower were transferred into two stages, the first one, is the transfer of heat and mass from bulk water temperature to an interface layer separated water from surrounding air, while the second stage is the transfer of heat and mass from interface to surrounding air. After Merkel, many theories and work were achieved. The heat and mass transfer equations of evaporative cooling in wet cooling towers were derivative by (Kloppers, 2005). He was derived the governing equations of the rigorous Poppe method of analysis from first principles. Also, the governing equations of the Merkel method of analysis were subsequently derived after some simplifying. A mathematical model based on heat and mass transfer principle was solved by (Saravanan *et al*, 2008) using an iterative method. The energy and exergy analysis showed that inlet air wet bulb temperature was the most important parameter than inlet water temperature. Also, variation in dead state properties does not affect the performance of wet cooling tower. (Ataei *et al*, 2008) had studied the thermal behavior of counter-flow wet cooling tower using a simulation model. The influence of the environmental conditions on the thermal efficiency of the cooling tower was investigated. The exergetic analysis was applied to study the cooling tower potential of performance improvement. (Bozorgan, 2010) presented a mathematical model based on the principle of heat and mass transfer between water and air. The module was used to analyze the energy and exergy of cooling towers in Khuzestan Steel Co.

The results showed that the exergy of water constantly decreases from the upper part of the tower to its lower part, while, the exergy of air was increased from bottom to top along the tower. A simple differential equation for counter flow wet cooling tower was solved analytically by (Yilmaz, 2010).

The obtained values were compared with the logarithmic mean enthalpy method (LMED) and corrected LMED method. It was found that the analytically obtained values were much more accurate than the values obtained using (LMED) or corrected LMETD methods. The performance of a wet-cooling tower fill using a model developed by Reuter was introduced by (Mehta *et al*, 2012); the model was derived in Cartesian coordinates for a rectangular cooling tower. The Reuter model was found effectively give the same results as the Poppe method for cross and counter flow fill configuration. A model for counter flow cooling tower was present by (Kotb, 2013). The Bosnjakovic formula and mutative water and air

*Corresponding author: **Abdul Hadi N. Khalifa**

properties were used to relax the constraints. The finite volumes of water and moist air were defined separately in opposite flow directions. The model was validated with experimental data from the literature. It was found that the height of the cooling tower was affected by the inlet air humidity.

In this work, Merkel theory for counter flow induced draught cooling tower was derived with some simplifications. The governing equation was solved by an iterative method. The effect of heat and mass transfer in counter flow cooling tower on cooling tower performance, exergy flow and exergy efficiency were introduced. The effect of the ratio of mass flow rate of water to that for air (L/G) on the tower performance and exergy destruction were also studied.

## 2. Thermal analysis of cooling tower

Many assumptions were made by Merkel to simplify the solution of cooling tower performance. Some of them were, saturated air leaving cooling tower, thus leaving air characterized by its enthalpy only, and mass flow rate of water through cooling tower is constant, i.e. neglected the reduction in mass flow rate due to evaporation of water (.Baker *et al*, 1961).

The mass balance of the cooling tower of the element shown in Fig. 1 is (Kloppers, 2003):

$$dL = G.d\omega \tag{1}$$

While the energy balance of the control volume is:

$$G.dh_a = L.c_w.T_w + c_w.T_w.dL \tag{2}$$

Substitute Eq. (1) into Eq. (2) and rearrangement the yields equation:

$$dT_w = \frac{G}{L}.\left(\frac{dh_a}{c_w} - T_w.d\omega\right) \tag{3}$$

In direct contact between two flow streams of water and air, like in counter flow cooling tower, the enthalpy potential is useful to find the amount of sensible and latent heat that exchange between them, the amount of total heat exchange can be written as, (Stoecker *et al*, 1982):

$$dq_t = dq_{lat.} + dq_{sen.} \tag{4}$$

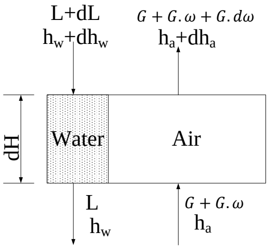


**Fig. 1** Exchange of mass and energy through cooling tower control volume

The latent heat transfer is due to the transfer of water vapour from the saturated air to moist stream air. Thus the equation of latent heat can be written as:

$$dq_{lat.} = h_{diff.}(\omega_s - \omega).h_g.dA \tag{5}$$

While The sensible heat is transferred due to the difference between water and air temperatures and can be written as:

$$dq_{sen.} = h_c.(T_w - T_a).dA \tag{6}$$

To change the temperature difference to enthalpy difference in Eq. (6), some information about the enthalpy of moist and saturated air should be used as follows: The enthalpy moist air can be written as follows (CIBSE Guide, 1986)

$$h_a = c_{pa}.T_a + \omega.\left(h_{fg} - c_{pw}.T_a\right) \tag{7a}$$

If the air is saturated at bulk water temperature then the enthalpy of saturated is:

$$h_{a,sat} = c_{pa}.T_w + \omega_{sat}.\left(h_{fg} - c_{pv}.T_w\right) \tag{7b}$$

Subtract Eq. (7a) from Eq. (7b) yields:

$$(h_{a,sat} - h_a) = \left(c_{pa} + \omega_{sat}.c_{pv}\right).T_w - (c_{pa} + \omega.c_{pv}).T_a + (\omega_{sat} - \omega).h_{fg} \tag{8}$$

Eq. (8) can be simplified by the following assumptions (Stoecker *et al*, 1982);

The difference between moisture content of saturated air and that for moist air is very small, thus, the first and second brackets in the right hand of Eq. (8) can be set as $c_{pm}$ which is given by the following equation:

$$c_{pm} = \left(c_{pa} + \omega.c_{pv}\right) = \left(c_{pa} + \omega_{sat}.c_{pv}\right) \tag{9}$$

Adding the term $(\omega_{sat} - \omega).h_f$ to Eq. (8) to change the enthalpy of evaporation of water $(h_{fg})$ to enthalpy of dry saturated water vapour $(h_g)$.

Using the assumptions above and rearrange Eq. (8) yields:

$$T_w - T_a = \frac{(h_{a,sat} - h_a) - (\omega_{sat} - \omega).h_g}{c_{pm}} \tag{10}$$

Substitute Eq. s (5), (6) and (10) into Eq. (4) yields:

$$dq_{t.} = h_{diff.}\left[\frac{h_c}{c_{pm}.h_{diff.}}.\left(h_{a,sat} - h_a\right) + \left(1 - \frac{h_c}{c_{pm}.h_{diff.}}\right).(\omega_{sat} - \omega).h_g\right].dA \tag{11}$$

By using the second assumption of Merkel, the second terms between brackets in the right hand of Eq. (11) can be neglected, yields:

$$dq_{t.} = \frac{h_c.dA}{c_{pm.}}.\left(h_{a,sat} - h_a\right) \quad (12)$$

The amount of total heat transferred through cooling tower also depends on the variation of water temperature, specific heat, and mass flow rate of water through cooling tower, thus, the total heat transferred through cooling tower can be written as:

$$dq_{t.} = L.c_w.dT \quad (13)$$

Equating Eq. s (12) and (13) yields:

$$L.c_w.dT = \frac{h_c.dA}{c_{pm.}}.\left(h_{a,sat} - h_a\right) \quad (14)$$

The integration of Eq. (14) gives the number of transfer unit of the cooling towers (NTU) (ASHRAE, 2008)

$$\frac{h_c.A}{c_{pm}} = = \frac{c_W.L.\Delta T}{(h_{a,sat}-h_a)_m} \quad (15)$$

The denominator in the hand right of the Eq. (15) $(h_{a,sat} - h_a)_m$ is the arithmetic enthalpy difference of increment element.

The effectiveness of cooling is the ratio between the actual to the maximum energy transfer through the tower:

$$\epsilon = \frac{h_{a,e}-h_{a,i}}{h_{a,sat}-h_{a,i}} \quad (16)$$

Lewis number for air water vapor systems can be written as (Bošnjaković, 1965);

$$Le_n = 0.865^{0.667}\frac{\left(\frac{\omega_S+0.662}{\omega+0.662}-1\right)}{\left(\frac{\omega_S+0.662}{\omega+0.662}\right)} \quad (17)$$

## 3. Exergy analysis of cooling tower

The exergy flow due to moist air consist of three terms, the first one is the thermal exergy, which can be calculated from the following equation:

$$\psi_{\text{the.}} = \left(c_{pa} + \omega.c_{pv}\right).T_o.\left(\frac{T_a}{T_o} - 1 - \ln\frac{T_a}{T_o}\right) \quad (18)$$

and the second term represent the chemical exergy:

$$\psi_{\text{Chem.}} = R_a.T_o\left[(1 + 1.68.\omega).\ln\frac{1+1.68.\omega_o}{1+1.68.\omega} + 1.68.\omega.\ln\frac{\omega}{\omega_o}\right] \quad (19)$$

While the third term is the mechanical exergy:

$$\psi_{\text{mech.}} = (1 + 1.68.\omega).R_a.\ln\left(\frac{P}{P_o}\right) \quad (20)$$

Since the pressure loss through cooling tower is insignificant factor in this work, so, the term of mechanical exergy can be neglected.

Thus, the exergy of moist air is the summation of thermal and chemical exeriges.

$$\psi_a = \psi_{\text{the.}} + \psi_{\text{Chem.}} \quad (21)$$

The water exergy is, (Dincer *et al*, 2012).

$$\psi_w = (h_w - h_o) - T_o.(s_w - s_o) \quad (22)$$

And the exergy to water vapour is,( Marletta, 2010) :

$$\psi_v = c_w.(T_w - T_o) - T_o.c_w.\ln\left(\frac{T_w}{T_o}\right) - T_o.c_w\ln(\phi_o) \quad (23)$$

The exergy destruction through element shown in Fig. 2 equals to the difference between the exergy in and exergy out of each of the followings; moist air, water and water vapour.

$$\psi_{dest} = \left[G.\left(\psi_{a,o} - \psi_{a,i}\right)\right] + \left[(L - G.\omega_o)\psi_{w,o} - (L - G.\omega_i)\psi_{w,i}\right] + \left[G.\left(\omega_o.\psi_{v,o} - \omega_i.\psi_{v,i}\right)\right] \quad (24)$$

The first term of the right hand of Eq. (24) is the difference between the outlet and inlet the exergy of air, the second term is the exergy difference between outlet and inlet water vapour, while the third one is the exergy difference between outlet and inlet water.

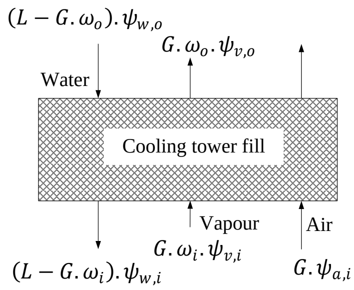


**Fig. 2** eEergy balance through cooling tower control volume

The exergy efficiency of any element through cooling tower is unity minus the ratio between the exergy destruction to the exergy inlet to the element (Bozorgan *et al*, 2012):

$$\eta_{exe.C.V} = 1 - \frac{\psi_{dest}}{\psi_i} \quad (25)$$

After a numerical integration for the governing equations, the cooling tower was divided into 100 horizontal elements, temperature difference in each element was 0.1 K. knowing outlet water temperature from tower bottom and inlet wet bulb temperature to upper section of tower., the thermal properties of air and water can be found using Engineering Equation Solver (EES) software.

## 4. Results and discussion

The performance of cooling tower was studied at inlet air dry and wet bulb temperatures of 50 and 25°C respectively, outlet water temperature is 32.5°C, volume flow rate of water is 200 $m^3$/hr and ratio of mass flow rate of water to that for air (L/G) is 1.25. Fig. 3 shows a path of saturation process through cooling tower on the psychrometric chart; this path is curvature toward the saturation line at a temperature equals inlet water temperature. The deviation of the processes from straight line is due to the assumption made by Merkel, that the outlet air is saturated, while the figure shows unsaturated air leaving tower (Kloppers *et al*, 2005). The figure also shows, a reduction in inlet water temperature and increasing in moisture content of the air. This is due to that, air temperature is more than water temperature, therefore sensible heat will transfer from air to water, while, the mass will transfer from water to air because the moisture content of saturated air is more than that for moist air. Fig. 4 indicates the same trends for air temperature and moisture content, while, the reduction in water temperature shown in the figure is due to evaporation an amount of water, thus the latent heat of evaporation extracted from the water itself.

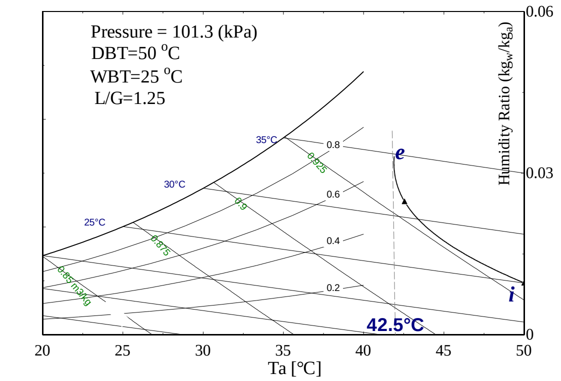


**Fig. 3** Saturation processes through cooling tower

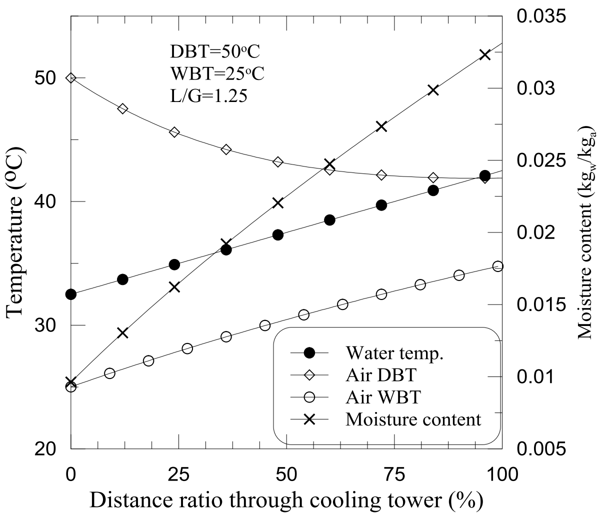


**Fig. 4** Temperature variation through cooling tower

Fig. 5 shows the variation of the number of transfer unit (NTU) and Lewis number through cooling tower, it can be seen from the figure that the NTU reduces along cooling tower due to increasing in enthalpy potential along cooling tower. Since the evaporation of water is neglected by Merkel, which is the main factor that affecting Lewis number. Therefore, Lewis number shows nearly constant trend through a cooling tower. Fig. 6 shows the variation of exergy efficiency and exergy destruction through a cooling tower. The exergy consumed in cooling tower is to destroy the exergy of hot water, so the more exergy destruction means lower water temperature is leaving the tower, or in the other word the higher exergy efficiency. Fig 7 shows the exergy flow of water as well as the exergy flow of air that consist of chemical and thermal exergy, it can be seen from the figure how the exergy of water destroyed from the top to the bottom of cooling tower, while the exergy of air increases from the bottom to the top of cooling tower. Since the mean factor affecting chemical exergy is the moisture content, therefore the chemical exergy increases through cooling tower as the moisture content increases. On the another hand, the thermal exergy tends to reduce along the tower due to convergence between air and water temperatures.

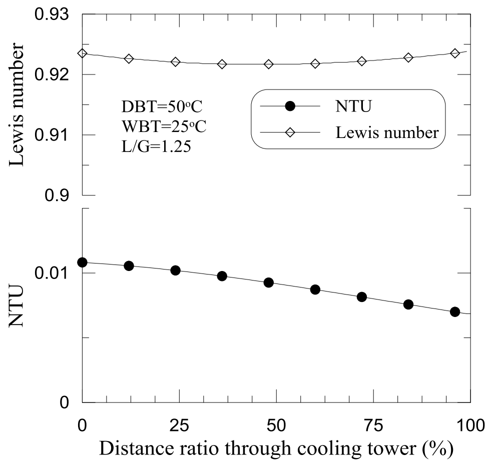


**Fig. 5** variation of NTU and Lewis number through cooling tower

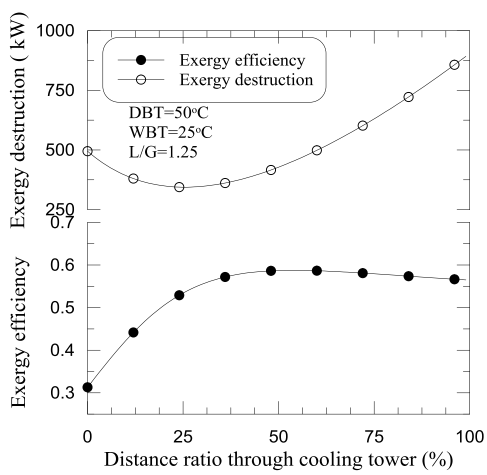


**Fig. 6** Exergy efficiency and exergy destruction through cooling tower

The effect of the ratio of mass flow rate of water to that for air (L/G) on the cooling tower performance was studied in the range of 0.75 to 2.25, for the conditions mentioned above. Fig 8 shows the variation of NTU with L/G ratio, it can be seen from the figure the NTU increases with the increasing of L/G ratio. The correlation for cooling characteristic tower curve (Baker *et al*, 1961) is NTU~(L/G)^n. The exponent n varies from about −0.35 to −1.1, the averages value between −0.55 and −0.65. If the value of (n) is taken as -0.6, the intersection point between such curve and NTU curve gives the L/G ratio of about 1.25 which is considered as the operating point of the cooling tower operates under the conditions mentioned above.

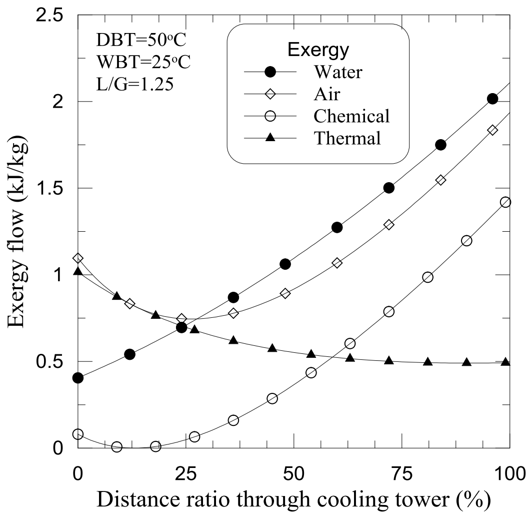


**Fig. 7** Exergy flow through cooling tower

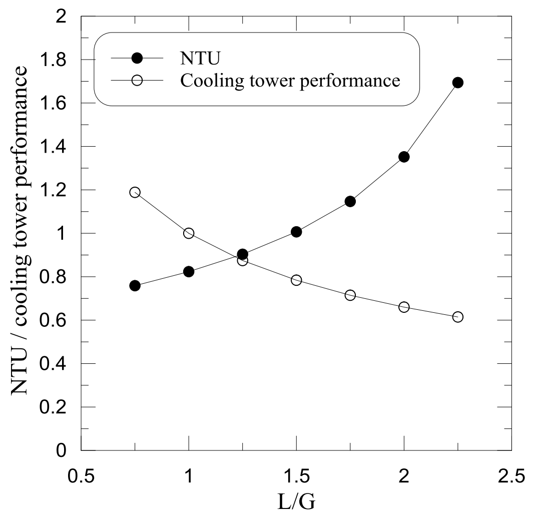


**Fig. 8** effect of (L/G) ratio on the NTU and tower performance

The exergy destruction increases with the increasing of L/G ratio, as it can be seen in Fig. 9, this increase is due to increase in entropy generation caused by convert liquid water to vapour. As mentioned previously, the more exergy destruction in cooling tower means a higher exergy efficiency, this can be indicated clearly in Fig. 10. The operating point of cooling tower related to exergy efficiency can be obtained from the same figure, by the intersection of exergy efficiency curve with that for cooling tower effusiveness, this intersection gives (L/G) ratio of about 1.5, and it is close to that obtained from Fig. 9.

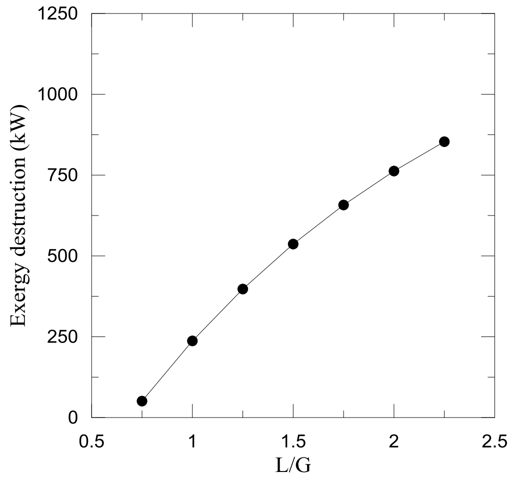


**Fig. 9** Exergy destruction vs (L/G) ratio and exergy efficiency

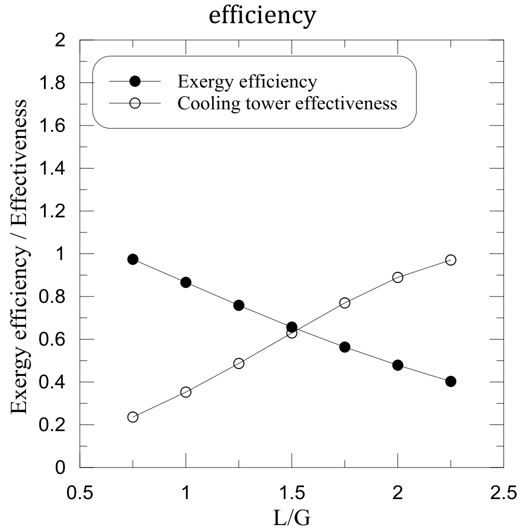


**Fig. 10** Effect of (L/G) ratio on cooling tower effectiveness

## Conclusions

1) Merkel assumption, which states that saturated air is leaving the cooling tower, gives curvature path for saturation process through cooling tower instead of a straight line
2) The ratio of mass flow rate of water to that for air (L/G) for such tower was in the range of 1.25 to 1.5.
3) In a cooling tower, the exergy of air is consumed to destroy the exergy of water, the more exergy destruction gives, the higher exergy efficiency for cooling tower.
4) As the moisture content of the air increases the air chemical exergy increases, on the another hand, as air temperature approaches to water temperature air thermal exergy tend to reduce.

## Nomenclature

| | |
|---|---|
| A | total area of wetted surface ($m^2$) |
| $c_{pa}$ | Specific heat of air at constant pressure (kJ/kg K) |
| $c_{pm}$ | specific heat of the air-water vapor mixture |
| $c_{pv}$ | Specific heat of water vapour at constant pressure (kJ/kg K) |
| $c_w$ | specific heat of water (kJ/kg K) |
| DBT | Dry bulb temperature of air ($^oC$, K) |
| G | mass flow rate of air through cooling tower (kg/s) |
| H | Enthalpy (kJ/kg), |
| $h_c$ | heat transfer coefficient ($W/m^2.K$) |
| $h_{diff.}$ | mass transfer coefficient ($kg/m^2s$) |
| L | mass flow rate of water through cooling tower (kg/s) |
| NTU | number of transfer unit |
| P | Pressure ( kPa.) |
| Q | heat transfer (kJ/kg) |
| $R_a$ | air gas constant (kJ/kg K) |
| S | entropy (kJ/kg K) |
| T | temperature ($^oC$, K) |

**Greek letters**

| | |
|---|---|
| η | Efficiency |
| Φ | Relative humidity of air |
| Ψ | Exergy flow (kJ/kg) |
| Ω | Moisture content ($kg_w/kg_{a)}$) |
| $\epsilon$ | cooling tower effectiveness |

**Subscripts**

| | |
|---|---|
| A | Air |
| chem. | Chemical |
| dest. | Destruction |
| E | Exit |
| Fg | Fluid-gas |
| G | Gas |
| I | In |
| lat. | Latent |
| M | Mean |
| mech. | Mechanical |
| O | Dead state |
| S | saturated |
| sen. | Sensible |
| T | Total |
| the. | Thermal |
| V | Vapour |
| W | Water |

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