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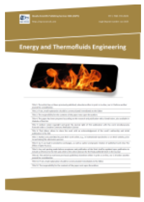

Numerical investigation of Merkel simplifying assumptions for the thermal analysis of forced draft wet cooling tower

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Abstract: The heat-mass transfer in the packing zone of a forced draft wet cooling tower is investigated in the present study by two different one-dimensional models like Merkel model and the Klimanek & Bialecki model. Merkel's model is based on Merkel simplifying assumptions, unlike Klimanek & Bialecki's model. After the ordinary differential equations, systems are derived for both the models, then, these equations systems are solved numerically by the Runge-Kutta method, to carry out, the water and the air temperatures, the humidity, and also other proprieties variation along the packing zone. Finally, after we have compared the results of both models with the same experimental data of a laboratory cooling tower. It has been noticed that both models are very accurate; however, the differences between the results of Merkel and Klimanek & Bialecki models with the experimental data at the packing outlet, are respectively about 0.14°C and 0.67°C for the water temperature, 0.07°C and 0.17°C for the air temperature. Furthermore, the relative humidity becomes 0.89% and 0.18% less in Merkel and Klimanek & Bialecki models respectively as compared to the experimental data for the same operating condition. Finally, it is noticed that the Merkel model is advantageous because of its easier equation derivation and is slightly more accurate than the Klimanek & Bialecki model.

1. Introduction

Evaporative cooling devices are widely used in the air conditioning field and also in many industrial and nuclear facilities. Recently, they have become more attractive because they can present an excellent solution to the high energy consumption and the environmental impact of the conventional cooling system. The cooling process of these devices can be achieved only through direct contact between air and water. During this contact, heat and mass transfer take place mainly due to the evaporation of a small quantity of water. This evaporation provides most of the cooling capacity of the evaporative cooling devices. Moreover, the humidity of the air increases, and when the air reaches its saturation line the cooling process will be interrupted. That may explain why the thermal performance of these kinds of devices is increased in hot and dry regions.

The modeling of the heat-mass transfer results by the direct air-water contact in the evaporative cooling system has been studied since the publication of the Merkel theory (Feltzin and Benton 1992). Among the many works that have been published; however, the present study is concerned with the work done by Alean et al. (2009). In his work, the author expressed the water and air temperatures, the humidity ratio, and other parameters by an ordinary differential equations system according to the Merkel simplifying assumptions. This system is solved numerically by the famous Range-Kutta method, to carry

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out, the change of water and air proprieties along the packing zone. To validate his results the author has made a comparison with experimental measurements which was obtained from a small-scale wet cooling tower. Klimanek & Bialecki (2009) and Meneceur et al. (2017) have studied the same heat and mass transfer problem in the wet cooling tower but without the simplifying assumptions done by Merkel. Both authors derived and solved numerically their mathematical models, and then the obtained results were validated by using elsewhere published results or experimental data.

Regarding the fact, that the draft wet cooling tower is largely used as the main component of many thermal and industrial systems, their thermal performance should be investigated and improved. The thermal performance of the draft wet cooling tower can be improved by optimizing its operational conditions, such as the inlet water temperature, the air and water mass flow rates, etc, further, by the arrangement and the type of packing, Lemouari et al. (2007), Asvapoositkul & Treeutok (2012), Rahmati et al. (2016), Shahali et al. (2016), Rahmati et al. (2018), Dmitriev et al. (2021), Yu et al. (2022), Xi et al. (2023), Wang et al. (2023).

Moreover, through the uses of nanofluids, Imani-Mofrad et al. (2016); Askari et al. (2016); Xie et al. (2017); Imani-Mofrad et al. (2018); Rahman et al. (2019); Elsaid (2020); Amini et al. (2020); Rahmati (2021); Hamid et al. (2021). However, the nanofluids use effects in the thermal performance of draft wet cooling towers are well illustrated by, Bakhtiyar et al. (2022) discussed the uses and the effect of three different compositions of MWCNT nanofluids type with 0.1 wt% of nono-additives concentration on the cross-flow wet cooling towers performance, such as, the evaporation rate, the performance characteristics, the temperature drop, and the tower efficiency. The obtained results show that the addition of nanoparticles even in small quantities to the water can improve the CFWCT performance by 15.8% compared to water. Javadpour (2022) has studied the effect of the two different types of nanofluid (MWCNTs and TiO_2) with a concentration of 0.05 wt%, furthermore, six kinds of filled beds on the performance of cross-flow wet cooling towers. The results revealed that the using of nanofluids instead of water generally improves the thermal performance of CFCT. Also, he shows how to make a suitable selection of the fill bed regarding the operating fluid either pure water or nanofluids. Continuing his previous work, Javadpour et al. (2021) also investigated the two nanofluids MWCNTs and TiO_2 effect on the thermal performance of a cross-flow cooling tower, such as the effectiveness, the Merkel number, and the cooling range. The obtained results reveal that for a concentration of 0.085 wt%, the Merkel number, the effectiveness, and cooling range can be improved by 28, 10.2, and 15.8%, respectively with the use of MWCNTs nanofluid, while they improved only by 5, 4.1, and 7.4%, for the TiO_2 nanofluid. He also finds out, that an optimal performance which about 23.5, 55.75%, and 0.64, respectively for the cooling range, effectiveness, and Merkel number, can be obtained for the MWCNTs nanofluid for a concentration of 0.069 wt% and a flow rate of 2.092 kg/min.

In this study, the heat-mass transfer problem is treated by two different models in the fill of a forced draft wet cooling tower. This cooling tower type is widely used in many air-conditioning and nuclear facilities, in order, to dissipate the heat from hot water to the atmosphere, through direct contact between water and upward airflow. During this contact, a heat and mass transfer takes place, as a result of water-air temperature difference and the evaporation of water as well. This heat-mass transfer problem is treated in this study, with the aim, of investigating the Merkel simplifying assumptions in the modeling of the heat-mass transfer in the cooling tower fill which has not been done before.

2. The Mathematical Modelling

The heat-mass transfer that occurred in the cooling tower fill, through direct contact between the hot water and the air, can be modeled by applying the mass and energy balances over a differential volume in the packing zone of the cooling tower, as shown in Fig. 1.

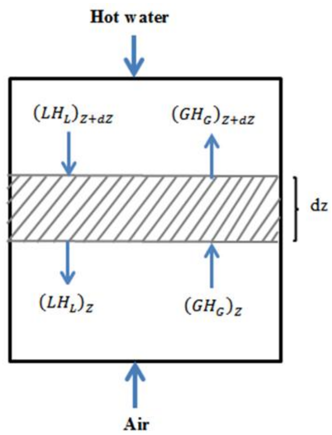

Fig. 1. A differential volume in the packing zone of the cooling tower.

The main mathematical formulas, further to the balances over the packing zone differential volume of mass and energy, which are used for the mathematical modeling and the derivation of the Merkel and Klimanek & Bialecki models, are presented below,

2.1. Enthalpy of Moist Air and Water Vapor

The enthalpy of moist air can be expressed as a function of the air temperature and the water's latent heat of evaporation, by the following formula (Kroger, 2004),

$$h_{ma} = C_{pa}T_a + w\left(C_{pv}T_a + H_{fg0}\right) \quad (1)$$

While, for saturation conditions, the enthalpy of moist air can be expressed by (Kroger, 2004),

$$h_{masw} = C_{pa}T_w + wh_v + h_v(w_{sw} - w) \quad (2)$$

The enthalpy of the water vapor, h_v at the bulk water temperature (Tw) is obtained as (Kroger, 2004):

$$h_v = C_{pv}T_w + H_{fg0} \quad (3)$$

The subtraction of the two Eqs. (1) and (2), give the Eq. (4), with, $C_s = C_{pa} + wC_{pv}$ is the specific heat of the moist air [J/kg°k].

$$h_{masw} - h_{ma} = C_s(T_w - T_a) + h_v(w_{sw} - w) \quad (4)$$

From the previous equation, we can also pull out,

$$(T_w - T_a) = \frac{1}{C_s}[(h_{masw} - h_{ma}) - h_v(w_{sw} - w)] \quad (5)$$

Where h_{ma}, h_{masw} $\left(\frac{J}{Kg}\right)$ are respectively, the enthalpy of the moist air and the enthalpy of moist air at saturation condition, C_{pa}, C_{pv} $\left(\frac{J}{Kg°K}\right)$ are respectively, the specific heat of the dry air and the water vapor, T_w, T_a $(°C)$ are respectively, the water and air temperatures, w is the humidity ratio of the moist air, and H_{fg0} $\left(\frac{J}{Kg}\right)$ is the latent heat of evaporation of water.

2.2. Mass Transfer to Air

The application of the mass balance in a differential volume Adz, reveals that the change in the water flow rate by evaporation is equal to the air humidification variation due to the mass transfer between water and air, Santos et al. (2011),

$$-d\dot{m}_w = \dot{m}_a dw = K_M a_M A(w_i - w)dz \quad (6)$$

where $\dot{m}_a, \dot{m}_w$ $\left(\frac{Kg}{s}\right)$ are respectively, the air and water mass flow rates, K_M $\left(\frac{\mathrm{kg}}{s\,m^2}\right)$ the mass transfer coefficient, w_i the air humidity ratio at the air-water interface, and a_M $\left(\frac{m^2}{m^3}\right)$ the available area to mass transfer per unit of volume of the packing.

2.3. Heat Transfer to Air

The sensible heat transfer between the air and air-water interface, leads to air temperature increases, as expressed in Eq. (7) (Santos et al., 2011),

$$\dot{m}_a C_s dT_a = h_a a_H A(T_i - T_a)\, dz \quad (7)$$

Where, h_a $\left(\frac{W}{m^2°K}\right)$ is the air heat transfer coefficient, a_H $\left(\frac{m^2}{m^3}\right)$ is the surface area available to heat transfer per unit of volume of the packing, and $T_i(°C)$ is the air temperature at the air-water interface.

2.4. Total Energy Transfer to Air

The total heat transfer between the air and air-water interface, due to the sensible and latent heats, is equal to the variation of the air enthalpy (Santos et al., 2011),

$$\dot{m}_a dh_{ma} = \dot{m}_a\left(C_s dT_a + H_{fg0}dw\right) = \left[K_M a_M A(w_i - w)H_{fg0} + h_a a_H A(T_i - T_a)\right]dz \quad (8)$$

After considering that the superficial areas a_M and a_H are identical and by involving the Merkel simplifying assumption of Lewis factor $(h_a/K_M C_s)$ which is considered to be equal to one. Then, Eq. (8) becomes (Santos et al., 2011),

$$\dot{m}_a dh_{ma} = K_M a_M A(H_i - H)dz \quad (9)$$

Where, h_{ma} (J/Kg), is the air enthalpy which is given by Eq. (1) and $(H_i - H)$ represents the enthalpy potential.

2.5. Energy Balance

According to the conservation of energy law for the differential volume Adz (Fig. 1), the rate of energy lost by the water is equal to the rate gained by the air (Kroger, 2004),

$$\dot{m}_a dh_{ma} = -\dot{m}_w C_{pw} dT_w - C_{pw} T_w d\dot{m}_w \quad (10)$$

2.6. Heat Transfer to Water

The decrease in the water temperature during the direct contact between water and air results from the heat exchange between the water and the water-air interface (Santos et al., 2011),

$$-\dot{m}_w C_{pw} dT_w = h_L a_H A (T_w - T_i) dz \tag{11}$$

3. Merkel Model

This model is derived according to the Merkel theory based on the following simplifying assumptions, which are as follows T. Hyhlık (2014):

- The Lewis factor relating heat and mass transfer is equal to 1.
- The air exiting the tower is saturated with water vapor and it is characterized only by its enthalpy.
- The reduction of water flow rate by evaporation is neglected in the energy balance.

The different ordinary differential equations of humidity, air, and water temperatures formed the Merkel model they are derived based on the previous equations, formulas, and simplifying assumptions as follows:

3.1. Humidity Ratio

The humidity ratio can be expressed from Eq. (6), by the ordinary differential equation (Klimanek & Bialecki, 2009),

$$\frac{dw}{dz} = \frac{K_M a_M A}{\dot{m}_a}(w_i - w) \tag{12}$$

3.2. Water Mass Flow Rate

The water mass flow rate variation due to the water evaporation can be evaluated by the following formula (Milosavljevic & Heikkilä, 2001):

$$\dot{m}_w(z) = \dot{m}_w - \dot{m}_a\big(w(z) - w(1)\big) \tag{13}$$

3.3. Sensible Heat

The sensible heat exchanged between the interface and the air is expressed from Eq. (8), by:

$$dq_s = h_a a_H A (T_i - T_a)\, dz \tag{14}$$

From the previous equation we can express the sensible heat by the ordinary differential equation (Alean et al., 2009):

$$\frac{dq_s}{dz} = h_a a_H A (T_i - T_a) \tag{15}$$

3.4. Latent Heat

The latent heat exchanged between the interface and the air can be also expressed from Eq. (8), by:

$$dq_l = K_M a_M A (w_i - w) H_{fg0} dz \tag{16}$$

From the last equation, the latent heat is expressed by the ordinary differential equation (Alean et al., 2009):

$$\frac{dq_l}{dz} = K_M a_M A H_{fg0}(w_i - w) \tag{17}$$

3.5. Air Temperature

To derive the air temperature differential equation, we start by differentiating the Eq. (1) with respect to (z), we obtain:

$$\frac{dh_{ma}}{dz} = C_{pa}\frac{dT_a}{dz} + wC_{pv}\frac{dT_a}{dz} + C_{pv}T_a\frac{dw}{dz} + H_{fg0}\frac{dw}{dz} \tag{18}$$

Now, we can pull out the derivative of T_a and after multiplying both sides of the result equation by the air mass flow rate we get:

$$\dot{m}_a C_s \frac{dT_a}{dz} = \dot{m}_a\frac{dh_{ma}}{dz} - \dot{m}_a\big(H_{fg0} + C_{pv}T_a\big)\frac{dw}{dz} \tag{19}$$

Thereafter, by substituting Eq. (8) into the previous one, we can finally obtain the ordinary differential equation of the air temperature (Alean et al., 2009).

$$\frac{dT_a}{dz} = \frac{[K_M a_M (w_i - w) H_{fg0} + h_a a_H (T_i - T_a)]A}{\dot{m}_a C_s} - \frac{\big(H_{fg0} + C_{pv}T_a\big)}{C_s}\frac{dw}{dz} \tag{20}$$

3.6. Water Temperature

From the energy balance over the control volume and after eliminating the small amount of the evaporated water, we can write (Kroger, 2004),

$$-\dot{m}_w C_{pw}\frac{dT_w}{dz} = \dot{m}_a\frac{dh_{ma}}{dz} \tag{21}$$

By replacing Eq. (8) with the precedent one we can get the following ordinary differential equation (Alean et al., 2009).

$$\frac{dT_w}{dz} = -\frac{\big[K_M a_M (w_i - w) H_{fg0} + h_a a_H (T_i - T_a)\big]A}{\dot{m}_w C_{pw}} \tag{22}$$

After we have considered that, $a_H = a_M = a$, $T_i = T_w$ and $w_i = w_{sw}$, all the ordinary differential equations from the Merkel model are presented in the equations system below (Alean et al., 2009).

$$\begin{cases} \dfrac{dw}{dZ} = \dfrac{h_a a\, A(w_{sw} - w)}{\dot{m}_a C_s} \\ \dfrac{dQ_s}{dZ} = h_a a\, A(T_w - T_a) \\ \dfrac{dQ_L}{dZ} = \dfrac{h_a a\, H_{fg0}\, A}{C_s}(w_{sw} - w) \\ \dfrac{dT_a}{dZ} = \dfrac{\left[h_a a(T_w - T_a) + \dfrac{h_a a}{C_s}(w_{sw} - w)H_{fg0}\right]A}{\dot{m}_a C_s} \\ \qquad\qquad - \dfrac{\left(H_{fg0} + C_{pv}T_a\right)}{C_s}\dfrac{dw}{dz} \\ \dfrac{dT_w}{dZ} = -\dfrac{\left[h_a a(T_w - T_a) + \dfrac{h_a a}{C_s}(w_{sw} - w)H_{fg0}\right]A}{\dot{m}_w C_{pw}} \end{cases} \quad (23)$$

To evaluate the saturated humidity ratio and the relative humidity of the air, the formulas represented below can be used (Kroger, 2004).

$$w_{sw} = 0.622\frac{P_{wvs}(T_L)}{P - P_{wvs}(T_L)} \quad (24)$$

$$\emptyset = \frac{wP}{(0.622 + w)P_{wvs}(T_a)} \quad (25)$$

To evaluate the atmospheric and the saturated water vapor pressures, mathematical formulas were used, see Appendix.

4. Klimanek and Bialecki Model

The ordinary differentials equations system that describe the water and the air properties variation along the packing zone are derived for this model as follows,

4.1. Humidity Ratio

The humidity ratio can be expressed directly from Eq. (6), by the ordinary differential equation (Klimanek & Bialecki, 2009),

$$\frac{dw}{dz} = \frac{K_M a_M A}{\dot{m}_a}(w_{sw} - w) \quad (26)$$

4.2. Water Mass Flow Rate

The mass balance over the packing zone differential volume yields to (Kroger, 2004),

$$\left(\dot{m}_w - \frac{d\dot{m}_w}{dz}dz\right) + \dot{m}_a(1 + w) = \dot{m}_w + \dot{m}_a\left(1 + w + \frac{dw}{dz}dz\right) \quad (27)$$

After simplification and by involving Eq. (26), we get the following differential equation (Kroger, 2004),

$$\frac{d\dot{m}_w}{dz} = -K_M a_M A(w_{sw} - w) \quad (28)$$

4.3. Convective Sensible Heat

Supposed that the interface temperature is equal to the water temperature, so the sensible heat exchanged between the interface and the air can be expressed from Eq. (8), by:

$$dQ_c = h_a a_H A(T_w - T_a)\, dz \quad (29)$$

4.4. Latent Heat

The latent heat exchanged between the interface and the air which results from the mass transfer between air and water is obtained from (Kroger, 2004):

$$dQ_m = h_v \frac{d\dot{m}_w}{dz}dz = h_v K_M a_M A(w_{sw} - w)dz \quad (30)$$

4.5. Total Heat

The total heat exchanged between the interface and the air can be obtained by the addition of the sensible heat and latent heat (Kroger, 2004),

$$dQ = dQ_c + dQ_m \quad (31)$$

After replacing the Eqs. (29) and (30) in the Eq.(31), we get,

$$dQ = [h_a a_H(T_w - T_a) + h_v K_M a_M(w_{sw} - w)]Adz \quad (32)$$

Now by substituting Eq. (5) and after considering that $a_M = a_H = a$ the equation of the total heat becomes (Kroger, 2004).

$$dQ = K_M a\, A(\left(1 - \frac{h_a}{K_M C_s}\right)h_v(w_{sw} - w) + \frac{h_a}{K_M C_s}(h_{masw} - h_{ma}))\, dz \quad (33)$$

By introducing the Lewis factor $\left(L_e = \frac{h_a}{K_M C_s}\right)$ in the Eq. (33) then we get,

$$dQ = K_M a\, A((1 - L_e)h_v(w_{sw} - w) + L_e(h_{masw} - h_{ma}))dz \quad (34)$$

On the other hand, it can also express the total heat transfer in the air-water interface as a function of the air enthalpy by:

$$Q = \dot{m}_a h_{ma} \quad (35)$$

After differencing the Eq. (35) with respect to (z) we find (Kroger, 2004):

$$\frac{dh_{ma}}{dz} = \frac{1}{\dot{m}_a}\frac{dQ}{dz} \tag{36}$$

Now by substituting Eq. (34) in Eq. (36), we get (Kroger, 2004):

$$\frac{dh_{ma}}{dz} = \frac{K_M a\, A}{\dot{m}_a}((1 - L_e)h_v(w_{sw} - w) + L_e(h_{masw} - h_{ma})) \tag{37}$$

4.6. Air temperature

To derive the air temperature differential equation we start by differentiating the Eq. (1) with respect to (z), we obtain:

$$\frac{dh_{ma}}{dz} = C_s\frac{dT_a}{dz} + \left(C_{pv}T_a + H_{fg0}\right)\frac{dw}{dz} \tag{38}$$

Now, after we pull out the derivative of T_a we get,

$$C_s\frac{dT_a}{dz} = \frac{dh_{ma}}{dz} - \left(C_{pv}T_a + H_{fg0}\right)\frac{dw}{dz} \tag{39}$$

After replacing the Eqs. (37) and (26) in Eq. (39) we get:

$$C_s\frac{dT_a}{dz} = \frac{K_M a\, A}{\dot{m}_a}((1 - L_e)h_v(w_{sw} - w) + L_e(h_{masw} - h_{ma}) - \left(C_{pv}T_a + H_{fg0}\right)(w_{sw} - w)) \tag{40}$$

Now by substituting Eq. (4) in Eq. (40) and after simplification we obtain,

$$\frac{dT_a}{dz} = \frac{K_M a\, A}{\dot{m}_a C_s}(h_v(w_{sw} - w) + L_e C_s(T_w - T_a) - \left(C_{pv}T_a + H_{fg0}\right)(w_{sw} - w)) \tag{41}$$

Now by involving Eq. (3), this leads to the following differential equation (Meneceur et al., 2017).

$$\frac{dT_a}{dz} = \frac{K_M a\, A}{\dot{m}_a C_s}(C_{pv}(T_w - T_a)(w_{sw} - w) + L_e C_s(T_w - T_a)) \tag{42}$$

4.7. Water Temperature

The energy balance over the packing zone differential volume, without neglecting the small amount of evaporation water can be expressed as follows (Kroger, 2004):

$$\dot{m}_a\frac{dh_{ma}}{dz} = -\dot{m}_w C_{pw}\frac{dT_w}{dz} - C_{pw}T_w\frac{d\dot{m}_w}{dz} \tag{43}$$

After pulling out the derivative of the water temperature from Eq. (43) and by involving firstly the Eqs. (37) and (28) and thereafter the Eqs. (4, 3), then after simplification the following differential equation is obtained (Meneceur et al., 2017).

$$\frac{dT_w}{dz} = -\frac{K_M a\, A}{\dot{m}_w C_{pw}}((C_{pv}T_w - C_{pw}T_w + H_{fg0})(w_{sw} - w) + L_e C_s(T_w - T_a)) \tag{44}$$

All the ordinary differential equations that form the Klimanek and Bialecki model are presented in the equations system below (Meneceur et al., 2017).

$$\begin{cases} \frac{dw}{dz} = \frac{K_M a_M A}{\dot{m}_a}(w_{sw} - w) \\ \frac{d\dot{m}_w}{dz} = -K_M a_M A(w_{sw} - w) \\ \frac{dQ_c}{dz} = h_a a_H A(T_w - T_a) \\ \frac{dQ_m}{dz} = h_v K_M a_M A(w_{sw} - w) \\ \frac{dT_a}{dz} = \frac{K_M a\, A}{\dot{m}_a C_s}\begin{bmatrix} C_{pv}(T_w - T_a)(w_{sw} - w) \\ +L_e C_s(T_w - T_a) \end{bmatrix} \\ \frac{dT_w}{dz} = -\frac{K_M a\, A}{\dot{m}_w C_{pw}}\begin{bmatrix} (C_{pv}T_w - C_{pw}T_w + H_{fg0})(w_{sw} - w) \\ +L_e C_s(T_w - T_a)) \end{bmatrix} \end{cases} \tag{45}$$

The Lewis number is evaluated by Bosnjakovic formula (Kroger, 2004),

$$L_e = \frac{h_a}{K_M C_s} = \frac{0.866^{0.667}\left(\frac{w_{sw} + 0.622}{w + 0.622} - 1\right)}{ln\left(\frac{w_{sw} + 0.622}{w + 0.622}\right)} \tag{46}$$

The air heat transfer coefficient is calculated from the Lewis factor formula,

$$h_a = K_M C_s L_e \tag{47}$$

5. Results and Discussions

To investigate which model is the more accurate, the equations systems of both models were solved numerically for the operating conditions and geometry data of a laboratory wet cooling tower which are provided in the Table. 1.

The main obtained results of both models and the comparison with the experimental data of the work, Alean et al. (2009) are given graphically in the figures below.

In Figs. 2 and 3, we show the water and the air temperature variation along the packing zone, obtained by both models, where is evident, that the water and air temperatures are inversely proportional because the amount of heat losses by the water is gained by the air.

Table 1. Operating Conditions and Geometry Data of the Cooling Tower (Alean et al., 2009).

parameters	values	parameters	values
Inlet air mass flow rate (kg/s)	0.0742	Wet bulb temperature (°C)	26.4
Inlet water mass flow rate (kg/s)	0.1262	the tower Area (m^2)	0.0625
Inlet air temperature (°C)	32	The packing height (m)	1.42
Inlet water temperature (°C)	40.8	Mass transfer coefficient (Kg/m^3s)	1.4
Inlet relative humidity (%)	65	Altitude of region (m)	169

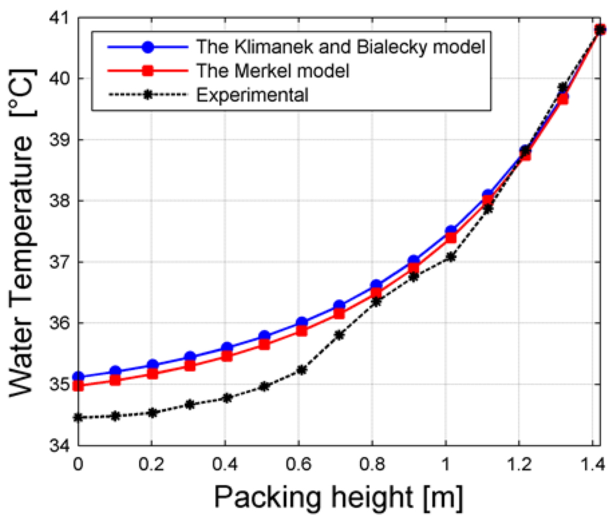

Fig. 2. The water temperature variation along the packing height.

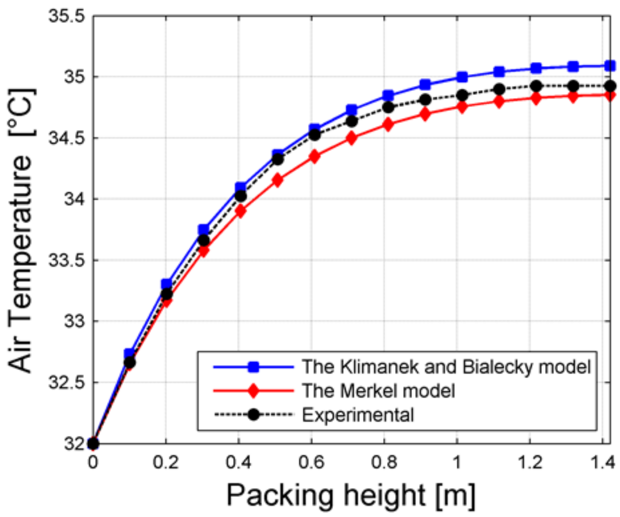

Fig. 3. The air temperature variation along the packing height.

In Fig. 4, we show the variation of the relative humidity along the packing zone obtained by both models where that is obviously; that the air is almost reaches the saturation at the exit of the packing, due to the evaporation of water during their contact with air.

In Fig. 5, we show the saturated humidity and humidity ratio variations along the packing zone, it's obvious that the humidity ratio increases as a result of mass transfer between water and air, and their variation is inversely proportional to the saturated humidity ratio, but they almost converge to the same value at the exit of the packing.

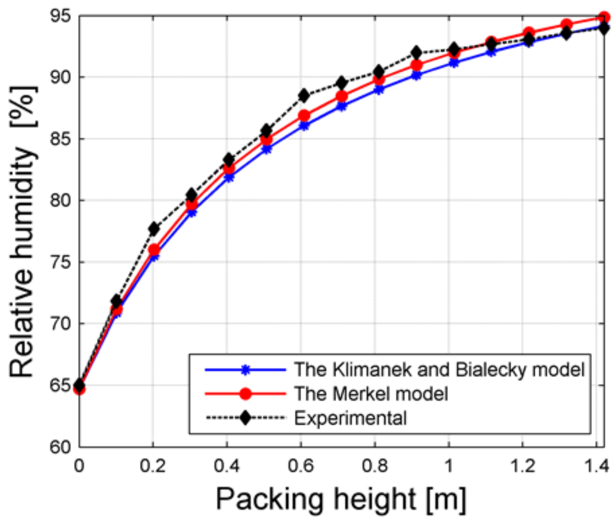

Fig. 4. The relative humidity variation along the packing height.

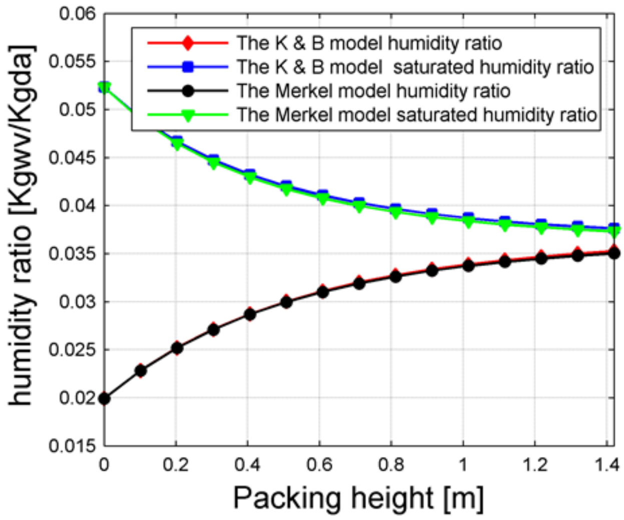

Fig. 5. The saturated and the humidity ratio variations along the packing height.

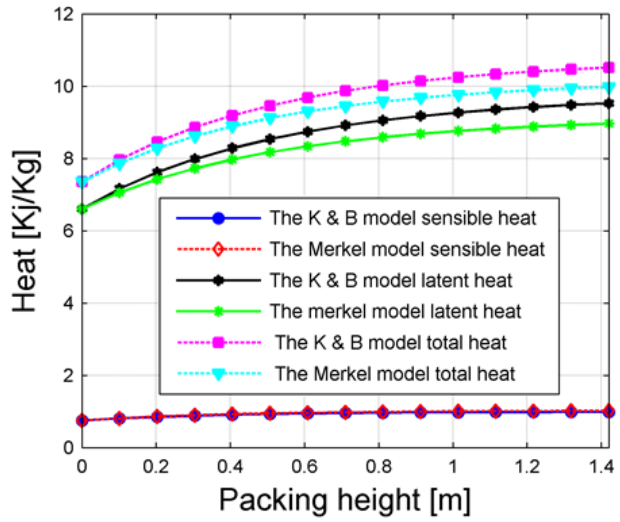

Fig. 6. The latent, sensible, and total heat variation along the packing height.

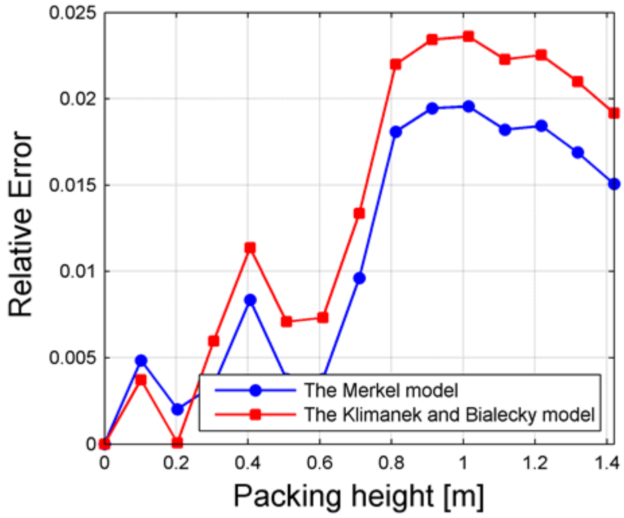

Fig. 8. The relative error of the water temperature for the Merkel and Klimanek & Bialecki models.

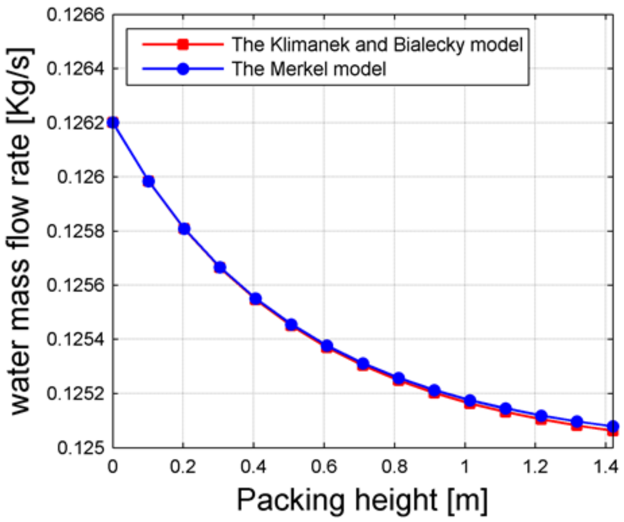

Fig. 7. The water mass flow rate variation along the packing height.

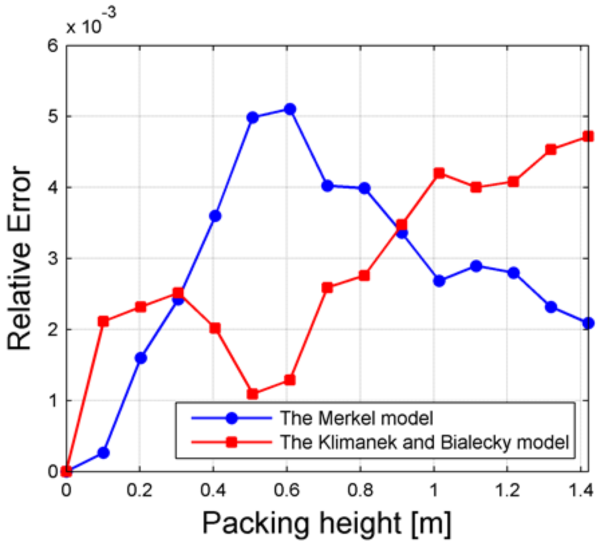

Fig. 9. The relative error of air temperature for the Merkel and Klimanek & Bialecki models.

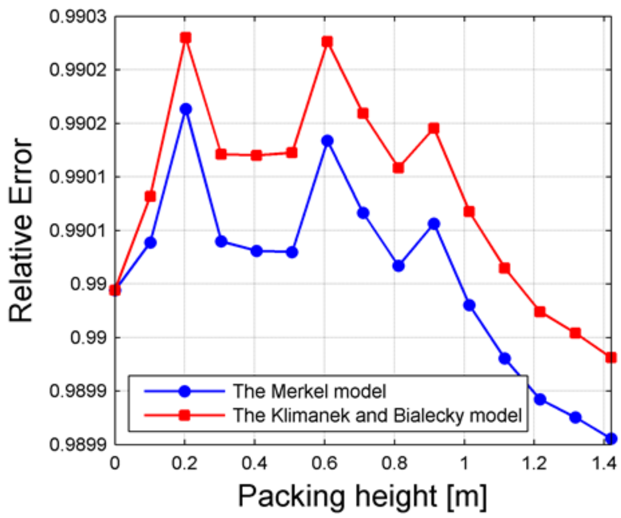

Fig. 10. The relative error of the relative humidity for the Merkel and Klimanek & Bialecki models.

In Fig. 6, we show the latent and sensible heat variations along the packing, where it's evident that the contribution of the latent heat in the cooling process is more significant than the sensible heat. This proves that the heat exchange between water and air is dominated by the evaporative heat transfer rather than the convective one.

In Fig. (7), we show the variation of the mass flow rate of the water along the packing, it's clear that the mass flow rate is decreased, due to, the evaporation of a small amount of water during the contact between water and air.

In Figs., 8, 9, and 10, we show the variation of the relative error which is calculated in function of the results of both models and the experimental data, respectively for the water temperature, air temperature, and relative humidity, where it's obvious that the relative error given by both models is very small, but the relative error of the Merkel model is the smallest one specifically for the water temperature and relative humidity, but for the air temperature, the relative error of both models are inversely proportional.

6. Conclusion

In order, to investigate the simplifying assumptions supposed by Merkel to deal with the heat and mass transfer problem in the fill of forced draft wet cooling tower. In this study the heat-mass transfer problem is treated by two different models, in the first one, the simplifying assumptions of Merkel are considered, unlike the second one. So, after we have solved the problem of the heat and mass transfer by both models, then the water and the air temperatures, the humidity, and other proprieties are calculated and compared with the same experimental data of a laboratory cooling tower, from the obtained results of both models we have noticed following points:

- The water temperature dropped exponentially from the inlet temperature of 40.8 °C until it became equal to 34.98 °C and 34.45 °C at the fill outlet, respectively for the Merkel and Klimanek & Bialecki models while it's about 35.12°C for the experimental results.
- The air temperature is raised exponentially from the inlet temperature of 32 °C until becomes equal to 34.85°C and 35.09°C at the fill outlet, respectively for the Merkel and Klimanek & Bialecki models while it's about 34.92°C for the experimental results.
- The same behavior as the air temperature is noticed for the relative humidity which is raised from 65% until it reached almost the saturation at the filling outlet, where it became equal to 94.85% and 94.14% respectively for the Merkel and the Klimanek & Bialecki models, while it's about 93.96% for the experimental results.
- The cooling process in the wet cooling towers is dominated by the evaporative heat transfer in comparison with the convective one.
- However, after the analysis of the results, it's obvious that both models are very accurate, but despite the simplifying assumptions, the Merkel model is slightly more accurate than the Klimanek & Bialecki model.

Disclosures

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Nomenclature

aM	Available area to mass transfer per unit volume of the packing, m^2m^{-3}
aH	Available area to heat transfer per unit volume of the packing, m2m-3
A	Cross area of the cooling tower, m^2
C_p	Specific heat, $jkg^{-1°}K^{-1}$
C_s	Specific heat of moist air, $jkg^{-1°}K^{-1}$
ha	air heat transfer coefficient, $wm^{-2°}K^{-1}$
hl	water heat transfer coefficient, wm-2°K-1
hma	Enthalpy of moist air, jkg^{-1}
hmasw	Enthalpy of moist air at the saturation, jkg-1
H	Specific enthalpy, jkg-1
Hi	Specific enthalpy in the air-water interface, jkg-1
Hfgo	Latent heat of evaporation of water, jkg-1
KM	mass transfer coefficient, $kgs^{-1}m^{-2}$
P	Pressure, Pa
T	Temperature, °C
q,Q	Heat flux, w
w	Humidity ratio, kg of water vapor. kg of dray air^{-1}

Greek symbols

ϕ	Humidity relative, %

Subscripts

a	air
i	air-water interface
l,m	latent
M	mass
sw	at saturation
s,c	sensible
v	vapor
w	water
wvs	saturated water vapor

Abbreviation

Le	Lewis factor

Appendix

To evaluate the atmospheric and the saturated water vapor pressures the below formulas were used.

-The atmospheric pressure formula, ASHRAE Fundamental 2017 (SI Edition), chapter 1:

$$P = 101325\left(1-(2.25577*10^{-5}*Z)\right)^{5.2559}$$

Z in [m] and P is obtained in [kPa].

-To evaluate the saturated water vapor we have used the Buck formula (OMNIcalculator, 2024):

$$P_{wvs} = 0.61121e^{\left(\left(18.678-\left(\frac{T}{234.5}\right)\right)*\left(\frac{T}{(257.14+T)}\right)\right)}$$

T in [°C] and P_{wvs} is obtained in [kPa].

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