International Journal of Thermal Sciences 49 (2010) 349–355

Contents lists available at ScienceDirect

International Journal of Thermal Sciences

journal homepage: www.elsevier.com/locate/ijts

Heat and mass transfer analysis of a wavy fin-and-tube heat exchanger under fully and partially wet surface conditions

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ARTICLE INFO

Article history:
Received 19 March 2009
Received in revised form
23 July 2009
Accepted 24 July 2009
Available online 20 August 2009

Keywords:
Heat exchanger
Fin
Evaporative cooling
Model
Mass transfer

ABSTRACT

In this study a mathematical model of heat and mass transfer performance of a wavy fin-and-tube heat exchanger under wet surface condition is presented. The heat exchanger is a counterflow heat exchanger in which humid air and liquid are flowing in opposite direction. A water film that causes evaporative cooling of the humid air is circulated on the humid air side. The heat and mass transfer equations are first derived for fully wet heat exchanger and then by defining a wettability parameter, these equations are obtained for partially wet heat exchanger. In modeling, values of Lewis number and wettability parameter are not necessarily specified as unity. The temperature distributions of humid air, liquid and water film, and relative humidity distribution of humid air are obtained numerically. The theoretical results are found to be in good agreement with the available experimental measurements.

1. Introduction

Heat dissipation is more effective if the cooling effect brought by water evaporation is applied to heat exchangers, such heat exchangers are fin-and-tube heat exchangers which are widely used in engineering applications like air-conditioning and refrigeration systems. Such engineering applications include cooling towers, evaporative coolers and evaporative condensers in which heat is dissipated from the hot fluid to the atmospheric air by direct or indirect contact through not only sensible heat exchange but also evaporative latent heat exchange. The cooling performance is greatly improved by the effect of water evaporation [1–6].

The principle of evaporative cooling is based on evaporating water into air stream. Mass transfer by evaporation and heat transfer are interconnected so that evaporation needs heat to occur. For this reason evaporation process will cool the air stream. The temperature of the evaporating water will reach equilibrium value at which water can cool a liquid flow in a recuperative heat exchanger.

Evaporative cooling systems are commonly used in countries where the climate is very hot and dry. The potential energy savings envisaged by replacing conventional refrigerated systems by evaporative systems is approximately 75% [7].

Evaporative air cooling can provide cooling and ventilation with minimal energy consumption using water as a working fluid and avoiding the use of ozone-destroying chlorofluorocarbons (CFC), hydrofluorocarbons (HFC) or hydrochlorofluorocarbons (HCFC). Technology is simple, functional and can be used in industrial and commercial applications, and consumes less than a quarter of the energy of refrigerative air-conditioning systems [8]. One way to cool an air stream is to inject water droplets by means of a spray into it, this results in cooler and humid air [9].

Due to the simultaneous heat and mass transfer in evaporative heat exchanger, the cooling process in evaporative heat exchanger is more complex than in sensible heat exchanger. The evaporative heat exchange between parallel plates was analysed based on simplified analysis model assuming Lewis number being unity and the water-vapor-saturation line being linear [10]. Moreover, the main deficiency of the model proposed by Maclaine-Cross and Banks [10] is that it is based on a very important condition that the evaporating water film is stationary and continuously replenished at its surface with water at the same temperature. Despite its above-mentioned deficiencies, the model by Maclaine-Cross and Banks [10] can be applied for easy and fast estimation of the maximum theoretically possible effectiveness of wet surface heat exchangers. The performance of evaluation method for the evaporative coolers in cylindrical or plate shapes was proposed by Chen et al. [11]. A short-cut method was developed by Stoichkov and Dimitrov [12] for calculating the effectiveness of wet surface crossflow plate heat exchanger of indirect evaporative cooling with

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1290-0729/$ – see front matter
doi:10.1016/j.ijthermalsci.2009.07.022

Nomenclature

A_f	wavy fin area, m^2
A'_f	plane fin area, m^2
A_p	area of tubes between fins, m^2
c_p	specific heat, J kg^{-1} K^{-1}
D	tube outer diameter, m
G	conductance, W K^{-1}
h_a	heat transfer coefficient, W m^{-2} K^{-1}
h_p	psychometric enthalpy of humid air, J kg^{-1}
h''_v	enthalpy of saturated vapor, J kg^{-1}
j	Colburn factor
k	mass transfer coefficient, mol s^{-1} m^{-2}
K_{Le}	Carrier–Lewis number
l_vo	heat of vaporization, J kg^{-1}
$\dot{m}$	mass flow, kg s^{-1}
M_a	air molar mass, kg mol^{-1}
Nu	Nusselt number
L	heat exchanger length, m
Le	Lewis number
p	total pressure of humid air, Pa
p_d	waviness amplitude, m
p_v	saturated vapor pressure, Pa
P_l	longitudinal tube pitch, m
Pr	Prandtl number
P_t	transverse tube pitch, m
Re	Reynolds number
s	fin spacing, m
Sc	Schmidt number
T	temperature, °C
W	width, m
x	absolute humidity, kg$_\mathrm{w}$kg$_\mathrm{a}^{-1}$
x_f	half wavy length, m
z	length coordinate, m
z^*	dimensionless length coordinate
z^*_d	dimensionless dry length coordinate
z^*_w	dimensionless wet length coordinate

Subscripts

a	air
l	liquid
s	water film surface
tot	total
v	vapor
w	water

Greek symbols

ϕ	heat flow rate, W
η	fin efficiency
κ	wettability parameter
λ	thermal conductivity, W m^{-1} K^{-1}
ν	kinematic viscosity, m^2 s^{-1}
ρ	density, kg m^{-3}

a more improved Maclaine-Cross and Banks model [10]. A model of an evaporative condenser by Peterson et al. [13] investigated the performance of an evaporative condenser in which the air flow and water film are flowing counter-current relative to each other. The temperature of the condensing refrigerant was constant. Measurements were made in field conditions. The accuracy of the measurement was fairly poor and for that reason there were quite big deviations between calculated and measured results. Differential equations for the temperatures, mass flow and humidity are presented by Finlay and Harris [14] for an evaporative cooler with evaporating water and air flowing counter-current to each other. Calculations are compared to measurements with good agreement between them. A comprehensive and complete description of the theoretical calculation of evaporative coolers and condensers was presented by Hellmann [15] where in principle all possible flow configurations were handled. The most detailed treatment was for a system where all three flows – air, evaporating water and liquid – were flowing concurrent. Hellmann [15] also pointed out the ill-conditioned behavior of the differential equations solution of the system at certain input values.

In the models mentioned above, the mass flow rate of spray water was always assumed to be sufficient large to achieve complete surface wetting. However the results of experimental study by Facao and Oliviera [16] showed that the incomplete wetting might occur with relatively small mass flow rate of spray water.

It can be noted that most of the theoretical and experimental studies found in literature were paid attention to heat transfer characteristics; research on mass transfer characteristics under partially wet surface condition is still limited. As a consequence, the objective of this study is to provide a detailed method for analysing the heat and mass transfer performance of fin-and-tube heat exchangers under partially wet surface conditions. Though the value of Lewis number is not necessarily equal to unity, heat and mass transfer analogy was still used to determine the mass transfer coefficient between the wet surface and the air by knowing the heat transfer coefficient.

The main objective of this work is to present a detailed model that is applicable to fully or partially surface wetting conditions with non unity Lewis number and which takes into account the effects of spray water evaporation and spray water temperature variation along the heat exchanger surface. In this work differential equations of temperature, mass flow and absolute humidity will be derived thoroughly for an evaporative cooler where air and evaporative water film are in concurrent flow and both streams are in counter-current flow to the liquid on the other side of the heat exchanger.

2. Mathematical model

The schematic diagram of the wet heat exchanger is presented in Fig. 1. The heat exchanger is a counter-current heat exchanger where liquid and humid air are flowing in opposite direction. On the humid air side, the water film is circulated parallel to humid air flow.

In modelling, the following assumptions are made:

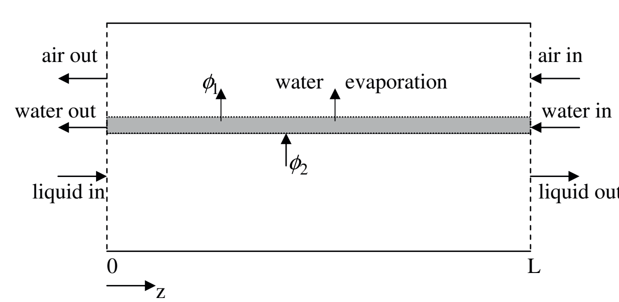

Fig. 1. Schematic diagram of heat exchanger.

- Liquid, water and air thermal properties are constant
- Temperature change of fluids is only in the direction of flow
- Wettability of the plate is uniform
- No heat transfer to the surroundings
- Constant heat and mass transfer coefficients
- Interface temperature is assumed to be the bulk water temperature

2.1. Fully wetted heat exchanger

The heat balance of the liquid flow can be written as

$$\dot{m}_l c_{pl} T_l(z) = \dot{m}_l c_{pl} T_l(z+\Delta z) + \phi_{1,\Delta z} \tag{1}$$

where $\dot{m}_l$ is the mass flow rate of liquid, $T_l(z)$ is the liquid temperature at z location and $T_l(z+\Delta z)$ at $z+\Delta z$ location and $\phi_{1,\Delta z}$ is the heat flow from liquid to the water film.

Considering the width of the heat exchanger to be W, we can write the heat flow from liquid to the water film as

$$\phi_{1,\Delta z} = G_1'' W \Delta z \left[\frac{1}{2}(T_l(z+\Delta z)+T_l(z)) - \frac{1}{2}(T_w(z+\Delta z)+T_w(z))\right] \tag{2}$$

where G_1'' is the conductance per unit area from the liquid to the water film and T_w is the water film temperature.

Combining Eqs. (1) and (2) and defining the dimensionless length $z^* = z/L (0 \le z^* \le 1)$, where L is the heat exchanger length, we get as $\Delta z \to 0$

$$\frac{dT_l}{dz^*} = \frac{G_1}{\dot{m}_l c_{pl}}(T_w - T_l) \tag{3}$$

where G_1 is the thermal conductance from the liquid to the water film.

The mass balance of the water film is

$$\dot{m}_w(z+\Delta z) + \dot{m}_{v,\Delta z} = \dot{m}_w(z) \tag{4}$$

where $\dot{m}_w$ is the water film mass flow and $\dot{m}_{v,\Delta z}$ is the evaporated mass flow from the water film to the humid air.

The surface of the water film is semipermeable, i.e. water vapor is diffusing through air but air is not diffusing through water. By means of mass transfer, the mass flow of water vapor per unit area can be expressed as

$$\dot{m}_v'' = kM_a\left[\frac{1}{2}(x_s(z+\Delta z)+x_s(z)) - \frac{1}{2}(x(z+\Delta z)+x(z))\right] \tag{5}$$

where x_s is the absolute humidity of the water film surface, x is the absolute humidity of the humid air, k is mass transfer coefficient and M_a is the molar mass of air.

Using Eqs. (4) and (5) and inserting $\dot{m}_v = \dot{m}_v'' W \Delta z$, we get as $\Delta z \to 0$

$$\frac{d\dot{m}_w}{dz^*} = kM_a WL\left(x_s\left(z^*\right) - x\left(z^*\right)\right) \tag{6}$$

Using heat and mass transfer analogy, the mass transfer coefficient can be obtained as

$$k = \frac{h_a}{c_{pa} M_a Le^{1-n}} \tag{7}$$

where h_a is the heat transfer coefficient between the water film surface and humid air, c_{pa} is the specific heat of air, Le is the Lewis number ($Le = Sc/Pr$) and $n = 0.42$ for turbulent flow.

Substituting Eq. (7) into Eq. (6), we get

$$\frac{d\dot{m}_w}{dz^*} = \frac{G_2}{c_{pa} Le^{1-n}}\left(x_s\left(z^*\right) - x\left(z^*\right)\right) \tag{8}$$

where G_2 is the thermal conductance from the water film surface to the humid air, which is expressed as

$$G_2 = h_a WL \tag{9}$$

The thermal energy balance of the flowing water film can be written as

$$\dot{m}_w c_{pw} T_w(z+\Delta z) + \phi_{1,\Delta z} = \phi_{2,\Delta z} + \dot{m}_w c_{pw} T_w(z) + \dot{m}_{v,\Delta z}\bar{h}_{v,\Delta z}'' \tag{10}$$

where c_{pw} is the specific heat of water, $\bar{h}_{v,\Delta z}''$ is the average enthalpy of saturated vapor within Δz and $\phi_{2,\Delta z}$ is the heat flow from the water film to the humid air.

The heat flow from the water film to the humid air can be obtained as

$$\phi_{2,\Delta z} = G_2'' W \Delta z \left[\frac{1}{2}(T_w(z+\Delta z)+T_w(z)) - \frac{1}{2}(T_a(z+\Delta z)+T_a(z))\right] \tag{11}$$

where T_a is the temperature of the humid air.

The average enthalpy of saturated vapor, $\bar{h}_{v,\Delta z}''$, is

$$\bar{h}_{v,\Delta z}'' = \frac{1}{2}\left(h_v''(z+\Delta z) + h_v''(z)\right) \tag{12}$$

Substituting Eqs. (2), (5), (7), (9), (11) and (12) into Eq. (10) and taking into consideration Eq. (8), we get as $\Delta z \to 0$

$$\begin{aligned}\frac{dT_w}{dz^*} = {} & \frac{G_2}{\dot{m}_w c_{pw}}(T_w - T_a) - \frac{G_1}{\dot{m}_w c_{pw}}(T_l - T_w) \\ & + \frac{G_2}{\dot{m}_w c_{pw} c_{pa} Le^{1-n}}(x_s - x)\left(h_v'' - c_{pw} T_w\right)\end{aligned} \tag{13}$$

The enthalpy of saturated vapor, h_v'', depends on the water film temperature, hence it depends on z^*, i.e. $h_v'' = h_v''(T_w(z^*))$.

The absolute humidity at the water film surface can be obtained from

$$x_s(T_w) = 0.622\frac{p_v(T_w)}{p - p_v(T_w)} \tag{14}$$

where p_v is the saturated vapor pressure at the water film surface and p is total pressure of the humid air which is the same as atmospheric pressure.

The mass balance of water vapor in the humid air is

$$\dot{m}_a x(z+\Delta z) + \dot{m}_{v,\Delta z} = \dot{m}_a x(z) \tag{15}$$

where $\dot{m}_a$ is the mass flow of dry air within the humid air.

Using Eqs. (5), (7) and (9), we get

$$\frac{dx}{dz^*} = \frac{G_2}{\dot{m}_a c_{pa} Le^{1-n}}(x - x_s) \tag{16}$$

The thermal energy balance of humid air is

$$\dot{m}_a h_p(z+\Delta z) + \dot{m}_{v,\Delta z}\bar{h}_{v,\Delta z}'' + \phi_{2,\Delta z} = \dot{m}_a h_p(z) \tag{17}$$

where h_p is the psychometric enthalpy of humid air.

The water vapor in the humid air is considered as an ideal gas, thus h_p of the humid air is

$$h_p = (c_{pa} + x c_{pv})T_a + x l_{vo} \tag{18}$$

where c_{pv} is the specific heat of water vapor, l_{vo} is the heat of vaporization of water at 0 °C and T_a is the humid air temperature in °C.

Combining Eqs. (5), (7), (11), (12), (17) and (18), we get

$$\frac{dT_a}{dz^*} = -\frac{G_2}{\dot{m}_a c_{pa}}(T_w - T_a) - \frac{G_2}{\dot{m}_a c_{pa}^2 Le^{1-n}}(x_s - x)\left(h_v'' - l_{vo}\right) \tag{19}$$

The change of mass flow of the water film is very small compared to the total mass flow of water film so the mass flow of water film is assumed constant hence Eq. (8) is discarded. Then Eq. (13) can be rewritten as

$$\frac{dT_w}{dz^*} = \frac{G_2}{\dot{m}_w c_{pw}}(T_w - T_a) - \frac{G_1}{\dot{m}_w c_{pw}}(T_l - T_w) + \frac{G_2}{\dot{m}_w c_{pw} c_{pa} Le^{1-n}}(x_s - x)h_v''. \tag{20}$$

2.2. Partially wetted heat exchanger

In the experimental work, the heat exchanger was sprayed with water by one nozzle placed in front of the heat exchanger. Using one nozzle is not likely enough to wet the whole surface of the heat exchanger, hence the heat exchanger is partially wetted. The distribution and the shape of the dry regions are very difficult to determine due to small spacing between the fins. Hence, it is assumed that the water is moving in narrow strings in the air flow direction.

Consider a small area with one part wet and one part dry, the dimensionless length of the wet part is dz_w^* and of the dry part is dz_d^*, then whole dimensionless length is

$$dz^* = dz_d^* + dz_w^* \tag{21}$$

and with the assumption that the wet length is distributed uniformly over the whole length, then we can write

$$\kappa = \frac{dz_w^*}{dz^*} \tag{22}$$

where κ is the wettability parameter $0 \le \kappa \le 1$.

Using Eqs. (21) and (22), we obtain

$$\frac{dz_d^*}{dz^*} = 1 - \kappa \tag{23}$$

Considering any quantity in Eqs. (3), (13), (16) or (19) and denoting it by f_w for the wet part and f_d for the dry part, we can then approximate the total behavior of f by

$$df = \frac{df_w}{dz_w^*}dz_w^* + \frac{df_d}{dz_d^*}dz_d^* \tag{24}$$

The function f can be any of the physical quantities of Eqs. (3), (13), (16) and (19).

Equation (24) can be rewritten as

$$\frac{df}{dz^*} = \frac{df_w}{dz_w^*}\frac{dz_w^*}{dz^*} + \frac{df_d}{dz_d^*}\frac{dz_d^*}{dz^*} \tag{25}$$

Since there is no water on the dry part, we obtain

$$\frac{dT_w}{dz_d^*} = 0 \tag{26}$$

The absolute humidity does not change when the air is flowing over the dry part, then we obtain

$$\frac{dx}{dz_d^*} = 0 \tag{27}$$

On the dry part of the heat exchanger, heat is transferred directly from the liquid through the heat exchanger wall to the humid air, hence there is no thermal resistance for the water layer.

For the dry part we get

$$\frac{dT_l}{dz_d^*} = \frac{G_{tot}}{\dot{m}_l c_{pl}}(T_a - T_l) \tag{28}$$

and

$$\frac{dT_a}{dz_d^*} = \frac{G_{tot}}{\dot{m}_a c_{pa}}(T_a - T_l) \tag{29}$$

where G_{tot} is the total conductance from the liquid to the humid air.

Combining all the relative equations according to Eq. (25), we get the final differential equations for the partially wetted heat exchanger as

$$\frac{dT_l}{dz^*} = \kappa\frac{G_1}{\dot{m}_l c_{pl}}(T_w - T_l) + (1-\kappa)\frac{G_{tot}}{\dot{m}_l c_{pl}}(T_a - T_l) \tag{30}$$

$$\frac{dT_w}{dz^*} = \kappa\left[\frac{G_2}{\dot{m}_w c_{pw}}(T_w - T_a) - \frac{G_1}{\dot{m}_w c_{pw}}(T_l - T_w) + \frac{G_2}{\dot{m}_w c_{pw} c_{pa} Le^{1-n}}(x_s - x)h_v''\right] \tag{31}$$

$$\frac{dT_a}{dz^*} = \kappa\left[-\frac{G_2}{\dot{m}_a c_{pa}}(T_w - T_a) - \frac{G_2}{\dot{m}_a c_{pa}^2 Le^{1-n}}(x_s - x)\left(h_v'' - l_{vo}\right)\right] + (1-\kappa)\frac{G_{tot}}{\dot{m}_a c_{pa}}(T_a - T_l) \tag{32}$$

$$\frac{dx}{dz^*} = \kappa\frac{G_2}{\dot{m}_a c_{pa} Le^{1-n}}(x - x_s) \tag{33}$$

For $\kappa = 1$, i.e. fully wetted heat exchanger, Eqs. (30–33) become exactly the same as those for fully wetted condition derived in Section 3.1.

2.3. Determination of thermal conductance

The total conductance can be determined by

$$\frac{1}{G_{tot}} = \frac{1}{G_1} + \frac{1}{G_2} \tag{34}$$

The conductance from liquid through the tube walls and plate fin to the surface of the water film G_1 consists of the conductance between the liquid and the inner tube surface G_l, conductance between inner and outer tube surfaces G_p and conductance between the outer tube surface and the base of the plate fin G_c, then we can write

$$\frac{1}{G_1} = \frac{1}{G_l} + \frac{1}{G_c} + \frac{1}{G_p} \tag{35}$$

Since the tube material has high thermal conductivity, the thermal resistance of the tube wall $1/G_p$ is small compared to the other thermal resistances, then

$$\frac{1}{G_1} = \frac{1}{G_l} + \frac{1}{G_c} \tag{36}$$

It is assumed that the fin surface temperature depends only on the spatial coordinate z, then the conductance G_2 is obtained as

$$G_2 = h_a\left(\eta A_f + A_p\right) \tag{37}$$

where h_a is the convective heat transfer coefficient on the air side, η is the fin efficiency, A_f is the area of the fins and A_p is the tubes area between the fins.

The heat transfer coefficient h_a depends on the air velocity between the fins and on the fin profile. For a herringbone wavy fin profile and staggered tubes arrangement, the heat transfer can be determined by using Colburn factor j [17]

$$j = 0.394 Re_D^{-0.357}\left(\frac{P_t}{P_l}\right)^{-0.272}\left(\frac{s}{D}\right)^{-0.205}\left(\frac{x_f}{p_d}\right)^{-0.558}\left(\frac{p_d}{s}\right)^{-0.133} \tag{38}$$

where Re_D is the Reynolds number, D is tube outer diameter, P_t is transverse tube pitch, P_l is longitudinal tube pitch, s is fin spacing, x_f is half wavy length and p_d is amplitude of waviness.

The heat transfer coefficient on the air side is obtained from

$$h_a = \lambda j Re_D Pr^{1/3}/D \tag{39}$$

where λ is the thermal conductivity of air.

The area of the wavy fins can be calculated by

$$A_f = \left[1 + \left(\frac{p_d}{x_f}\right)^2\right] A_f' \tag{40}$$

where A_f' is the area of the plane fins, i.e. non-wavy profile.

3. Experimental setup

The schematic diagram of the experimental assembly is shown in Fig. 2. It consists of a closed-loop of circulating water, nozzle, water tank, pumps and heat exchanger. The heat exchanger in the experimental assembly is a wavy plate fin-and-tube heat exchanger with the following data:

- Plates thickness is 0.3 mm, plates height is 600 mm and plates length (in the direction of air flow) is 290 mm

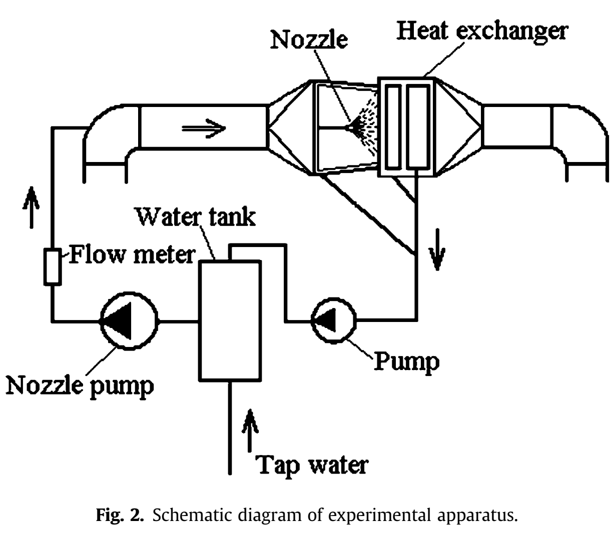

Fig. 2. Schematic diagram of experimental apparatus.

- Distance between the plates is 3 mm
- Number of plates is 200, made of aluminum ASTM 110
- Tube inner and outer diameters are 10 mm and 12 mm, respectively

The tubes are made of copper and they are placed in staggered arrangement with a horizontal distance of 32 mm between the centerlines of the tubes, longitudinal tube pitch is 29.4 mm and transverse tube pitch is 34 mm. The number of the tubes in one row is 18 and the number of rows is 10.

The herringbone wavy configuration of plate fins has the following dimensions:

- Fin spacing is 3 mm
- Wavy length is 8.5 mm
- Amplitude of waviness is 1 mm

The spraying system consists of 100 L water tank, pump and nozzle. The mass flow of the spraying water is 0.6714 kg s^{-3} which is measured by a rotameter. Tap water is used to fill the tank since the amount of water decreases as evaporation occurs. The nozzle is placed in front of the heat exchanger at 34 cm away from the front area of the heat exchanger and water is sprayed onto the heat exchanger in the direction of the air flow.

The working medium for the tube side is a water–potassium solution with a mass fraction of 34% potassium. The temperature of the circulating liquid in the heat exchanger is regulated by three-way control valve and by-pass pipe. The temperatures of the liquid at inlet and exit of the heat exchanger are measured by thermistors. The mass flow of the liquid leaving the heat exchanger is measured by a magnetic-inductive flow meter. The temperatures of the air flow at the inlet and exit of the heat exchanger are measured by thermistors. The position of thermistors in the air flow is determined and checked so that the heat balance of the heat exchanger in dry conditions was within 1% error.

4. Results and discussion

The thermal properties of the liquid (water–potassium solution) flowing in the tubes of the heat exchanger are: specific heat $c_p = 3.1$ kJ kg^{-1} K^{-1}, density $\rho = 1210$ kg m^{-3}, thermal conductivity $\lambda = 0.525$ W m^{-1} K^{-1} and kinematic viscosity $\nu = 1.2\acute{}\ 10^{-6}$ m^2 s^{-1}.

Knowing the liquid mass flow in the tubes $\dot{m}_l = 0.375$ kgs^{-1}, the Reynolds number of flowing liquid inside the tubes is calculated as $Re = 16{,}485$ hence the flow is turbulent. Then we obtain the liquid heat transfer coefficient as $h_l = 5439$ W m^{-2} K^{-1}. Multiplying h_l by the total inner area of tubes (3.72 m^2), we obtain the conductance as $G_l = 20{,}233$ W K^{-1}. The contact conductance is calculated as $G_c = 6700$ W K^{-1} based on completely dry heat exchanger. Using Eq. (36), we obtain the conductance $G_1 = 5033$ W K^{-1}. The air mass flow on the outer side of the heat exchanger is $\dot{m}_a = 0.902$ kgs^{-1} and knowing that the minimum flow area between the fins is 0.217 m^2, then the maximum air velocity is obtained as $u_{max} = 3.51$ m s^{-1}, from which we obtain the Reynolds number that is based on the maximum velocity as $Re_D = 2886$. Using Eq. (38), the Colburn factor j is calculated as $j = 0.0153$. Then using Eq. (39), the heat transfer coefficient on the air side is obtained as $h_a = 80.6$ W m^{-2} K^{-1}. Knowing the air side heat transfer coefficient, the fin efficiency is obtained as $\eta = 0.83$ according to Ryti [18]. The heat transfer area of non-wavy fins is $A_f' = 61.5$ m^2 then the area of wavy fins is calculated from Eq. (40) as $A_f = 63.1$ m^2. The area of the tubes between the fins is $A_p = 4.2$ m^2. Hence, the spray flow rate per tube surface area is 0.16 kg m^2 s^{-1} Using Eq. (37), the conductance G_2 is calculated as $G_2 = 4560$ W K^{-1}. Then the total conductance can be obtained from

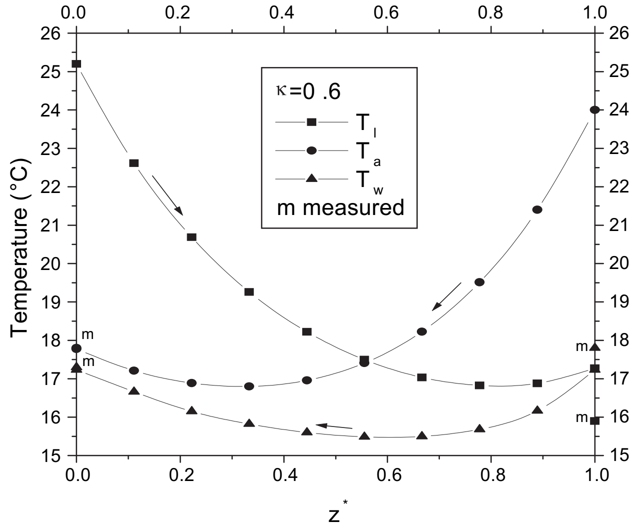

Fig. 3. Temperature profiles along axial direction for $\kappa = 0.6$.

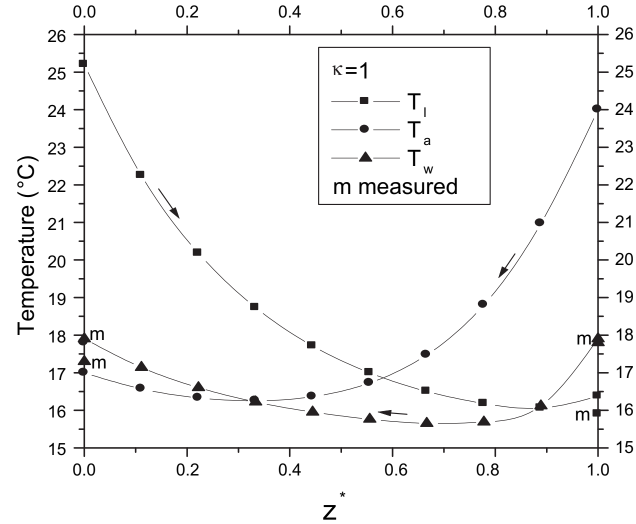

Fig. 5. Temperature profiles along axial direction for $\kappa = 1$.

Eq (34) as $G_{tot} = 2392\ \mathrm{W\,K^{-1}}$. The water mass flow is $\dot{m}_w = 0.667\ \mathrm{kgs^{-1}}$ and the Lewis number is $Le = 0.866$. The boundary conditions required to solve the differential equations (30–33) are: $T_l(0) = 25.2\ ^\circ\mathrm{C}$, $T_a(1) = 24\ ^\circ\mathrm{C}$, $T_w(0) = T_w(1)$ and $x(1) = 0.004918\ \mathrm{kg_w kg_a^{-1}}$.

For recirculation, inlet spray water temperature will be equal to its outlet value as can be seen from the water flow boundary condition.

The measured values from the experimental setup are: $T_l(1) = 15.9\ ^\circ\mathrm{C}$, $T_a(0) = 17.8\ ^\circ\mathrm{C}$, $T_w(0) = 17.3\ ^\circ\mathrm{C}$, $T_w(1) = 17.8\ ^\circ\mathrm{C}$ and $x(0) = 0.01265\ \mathrm{kg_w kg_a^{-1}}$.

As we see from the experimental results that the inlet water temperature slightly differ from its outlet value because the outlet water is circulated through a water tank before it enters the heat exchanger. The differential equations (30–33) along with the boundary conditions are solved numerically by using Runge–Kutta method. Figs. 3–5 show the change of liquid temperature (T_l), air temperature (T_a) and water temperature (T_w) along the flow direction for different wettability parameter ($\kappa = 0.6$, 0.8 and 1). It can be seen from Figs. 3–5 that the liquid temperature and the air temperature decrease in their flow direction. However the water temperature decreases from its inlet temperature and then increases to reach again its inlet temperature as can be seen from the water temperature boundary condition used in the model. With $\kappa = 0.6$, the measured exit air and water temperatures coincide with those measured values while exit liquid and inlet water temperature slightly deviate from their measured values as can be seen from Fig. 3. However as the wettability parameter κ increases, the exit liquid and inlet water temperatures approach their measured values, and the measured exit air and water temperatures slightly deviate from their measured values. Hence, the best theoretical temperatures compared to the measured values are obtained for $\kappa = 1$. Fig. 6 shows the variation of absolute humidity with air flow direction for different wettability parameter. The figure shows that the absolute humidity is increasing in the flow direction. As it is expected, Fig. 6 shows that the absolute humidity increases as the wettability parameter increases. Fig. 6 also shows that the measured exit air absolute humidity agrees most with the one obtained numerically for $\kappa = 1$. In general, the measured results agree very well with the values obtained numerically for $\kappa = 1$, which means that the water mass flow rate used in the analysis assures that the heat exchanger is completely wet, hence the spraying system used in the experimental setup is enough for the wetting condition.

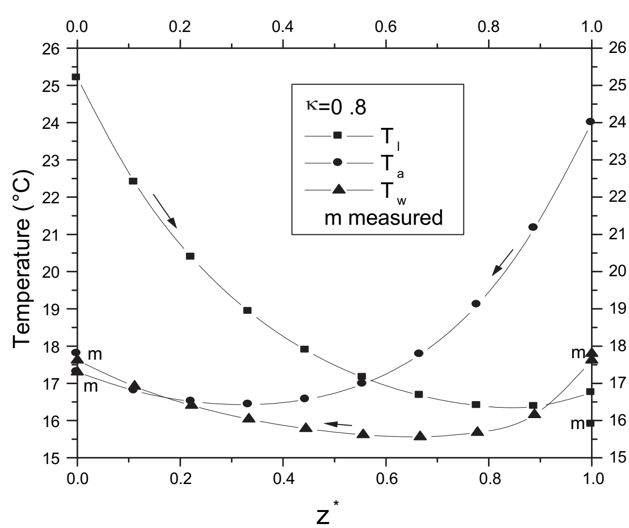

Fig. 4. Temperature profiles along axial direction for $\kappa = 0.8$.

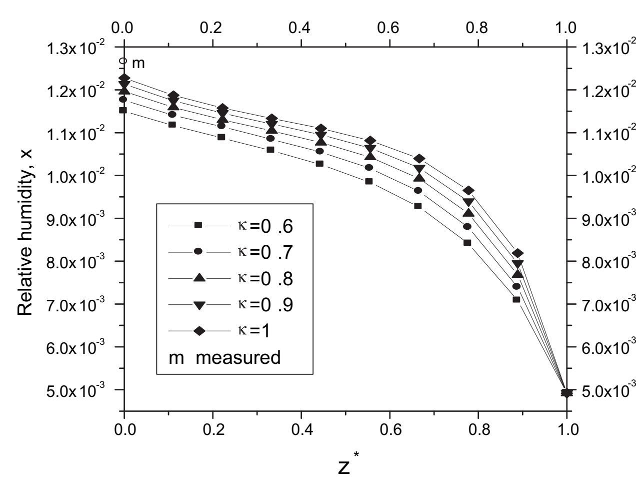

Fig. 6. Absolute humidity profiles along axial direction for different κ values.

5. Conclusions

This study theoretically and experimentally examined the heat and mass transfer characteristics of a fin-and-tube heat exchanger having wavy fin geometry. The mathematical model described heat and mass transfer processes in evaporative cooling systems for fully and partially wet surface conditions. The model was presented by a set of differential equations with boundary conditions for heat and mass transfer. These equations were solved numerically by means of finite-difference method. A comparison of calculations with the data obtained from full-scale experiment has shown that they coincide within tolerable measurement accuracy. The model can be used to reduce the experimental work due to its good accuracy.

Since the wettability parameter can be only determined through comparison with experimental results, more research will be conducted to relate the wettability parameter with the spray water flow rate. Moreover this model will be used in future research to optimize system configuration, e.g., diameter of tube and height/density of fins, as well as to obtain the suitable operating conditions, e.g., liquid flow rate, temperature, air flow rate and humidity.

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