Accepted Manuscript

The numerical model for direct evaporative cooler

Igor Kovačević, Maarten Sourbron

PII: S1359-4311(16)33025-3

DOI: <http://dx.doi.org/10.1016/j.applthermaleng.2016.11.025>

Reference: ATE 9435

To appear in: *Applied Thermal Engineering*

Received Date: 28 July 2016

Accepted Date: 3 November 2016

Please cite this article as: I. Kovačević, M. Sourbron, The numerical model for direct evaporative cooler, *Applied Thermal Engineering* (2016), doi: <http://dx.doi.org/10.1016/j.applthermaleng.2016.11.025>

This is a PDF file of an unedited manuscript that has been accepted for publication. As a service to our customers we are providing this early version of the manuscript. The manuscript will undergo copyediting, typesetting, and review of the resulting proof before it is published in its final form. Please note that during the production process errors may be discovered which could affect the content, and all legal disclaimers that apply to the journal pertain.

THE NUMERICAL MODEL FOR DIRECT EVAPORATIVE COOLER

Igor Kovačević*, Maarten Sourbron

November 4, 2016

KU Leuven, Department of Mechanical Engineering
Division of Applied Mechanics and Energy Conversion
Celestijnenlaan 300, B 2421
3001 Leuven, Belgium

ABSTRACT

This paper presents a numerical model for a compact direct-contact cross-flow air/water heat exchanger where evaporating water cools down an air stream, and where an innovatively designed metallic direct evaporative pad enhances air-water interaction. The numerical model implements energy and mass conservation equations of humid air and water in a one-dimensional geometry by applying correlations for heat and mass transfer coefficients. The system of ordinary differential equations is solved by central-finite discretisation using Matlab. The effective hydraulic diameter is isolated as the only unknown model parameter, and is determined by a parameter estimation using experimental data available from a producer of such a direct evaporative pad. The numerical model is able to predict the air outlet temperature, with an maximal error of 1.33 % compared to experimental data for different inlet temperature and humidity values. Humid air properties inside and at the outlet of the direct evaporative pad, the pad effectiveness and the water consumption can be evaluated by the presented model. The use of the numerical model is demonstrated with examples analysing the impact on heat exchanger effectiveness of a changed geometry (design analysis) and of varying air inlet conditions for a given geometry (operational analysis).

Keywords

Direct Evaporative Cooler, Numerical Model, Moist Air, Pad Effectiveness

*Corresponding author: igor.kovacevic@kuleuven.be

Highlights

- The numerical model for metallic-compact air/water direct evaporative cooler is developed;
- The effective hydraulic diameter is determined by a parameter estimation using experimental data available from a producer of the direct evaporative pad;
- Temperature and humidity of air throughout the cooler and pad effectiveness can be simulated by the developed numerical model;

Contents

1	Introduction	1
2	Physical Model	3
2.1	Governing Equations	4
2.2	Fluid Flow between Parallel Plates	6
2.3	Thermodynamic and Thermophysical Properties	9
3	Model Validation	9
3.1	Direct Evaporative Cooler	9
3.2	Description of Implemented Model	10
3.3	Numerical Calibration	11
3.4	Determination of Effective Hydraulic Diameter	12
3.5	Comparison between Numerical Results and Measurement Data	13
4	Model Results	14
4.1	Heat and Mass Transfer Coefficients	14
4.2	Temperature and Humidity Profiles	17
5	Conclusion	21

List of Figures

1	Schematic presentations of evaporative cooler and the physical model	3
2	Nusselt number in the entrance region between parallel plates	8
3	The Oxyvap cooler - the DEP used for model validation	10
4	The mass transfer coefficient profile used in numerical computation	16
5	The heat transfer coefficient profile across the first fifth fins on the layer	16
6	Heat transfer coefficients throughout the pad with different number of longitudinal fins . .	17
7	Temperature and humidity with variation of input air velocity. The input air properties are: $T_{in} = 34\text{ }^{\circ}\text{C}$ and $\varphi_{in} = 40\%$	19
8	Temperature and relative humidity with variation of the input relative humidity. Examples are given for the input air temperature $T_{in} = 30\text{ }^{\circ}\text{C}$ and the input air velocity $v = 2\text{ m/s}$.	19
9	Temperature and relative humidity with variation of input air temperature. 30 % is the input relative humidity and 2 m/s is the input air velocity	20

List of Tables

1	DEP effectiveness, CPU time and errors between numerical results ([%]) obtained by dif- ferent number of nodes for discretisation	11
2	Numerical computation of optimal hydraulic diameter in the model	13
3	Comparison of outlet temperatures computed by the developed numerical model and mea- surement data available ($v_{in} = 1\text{ m/s}$ and $\tilde{D}_h = 3.40\text{ mm}$).	14
4	The entrance length and the thickness of boundary layer at the end of longitudinal fin in variation of input air velocities	14
5	Variation of the heat and mass transfer coefficients and the DEP effectiveness with variation of the number of fins on layers in the same pad thickness	17
6	Output results of presented examples	20

Used Symbols

A , [m²] - area;
 A_f , [m²] - frontal area of the cooler;
 c_p , [kJ/kgK] - specific heat;
 D , [m] - distance between parallel plates;
 D_h , [m] - hydraulic diameter;
 $\tilde{D}$, [m] - particular distance between parallel fins;
 $\tilde{D}_h$, [m] - particular hydraulic diameter;
 D_{va} , [m²/s] - diffusivity of water vapour in air;
 h , [kJ/kg] - enthalpy;
 h_{fg} , [kJ/kg] - heat of evaporation;
 h_M , [m²/s] - mass transfer coefficient;
 $\bar{h}_M$, [m²/s] - average mass transfer coefficient;
 h_T , [W/m²K] - heat transfer coefficient;
 $\bar{h}_T$, [W/m²K] - average heat transfer coefficient;
 k , [W/mK] - conductivity;
 L , [m] - pad thickness/length;
 L_f , [m] - length of longitudinal fin;
 m , [kg] - mass;
 $\dot{m}$, [kg/s] - mass flow;
 $\Delta \dot{m}_w$, [kg/s] - water consumption;
 N , [–] - number of nodes for discretisation;
 $\bar{N}$ - reference number of nodes;
 N_f , [–] - number of fins in the pad;
 P , [m] - perimeter/width;
 $\dot{Q}_S$, [kJ] - sensible heat;
 $\dot{Q}_L$, [kJ] - latent heat;
 T , [°C] - temperature;

T_{wb} , [°C] - wet-bulb temperature;

v_{in} , [m/s] - input air velocity;

U , [m/s] - mean velocity;

w , [kg_v/kg_a] - absolute humidity;

w_s , [kg_v/kg_a] - saturated air humidity;

x , [m] - longitudinal coordinate;

x_* , [-] - longitudinal dimensional coordinate;

X , [m] - entrance length.

Dimensionless Number

Nu - Nusselt number;

Pr - Prandtl number;

Re - Reynolds number;

Sc - Schmidt number;

Sh - Sherwood number;

Greek letters

α , [m²/s] - thermal diffusivity;

β , [m²/m³] - density area;

δ , [m] - boundary layer thickness;

δ_f , [m] - thickness of layer/fin;

Δ , [-] - sum of differences;

ε , [%] - DEP effectiveness;

φ , [%] - relative humidity;

ρ , [kg/m³] - density;

ν , [m²/s] - kinetic viscosity;

ζ , [%] - difference between results;

Subscripts

a - air;

in - input value;

m - mixture of water vapour and air, humid air;

out - output value;

v - water vapour;

w - water;

Superscripts

e - measurement/experimental data;

m - modeling result;

1 Introduction

Evaporative cooling is a well-known, environmentally friendly technology with many useful applications. It can be used in residential buildings, air handling units for commercial/industrial buildings and agriculture. In some parts of the World, depending of climate conditions, this technology can be used for air conditioning [1] directly without using vapour-compressor cycle. Moreover, in vapour-compression/refrigeration systems this technology is applied as pre-cooling in front of an air-cooled condenser.

Xuan et al. [2] present research and application of evaporative cooling for air conditioning in China, where different wetted media and heat and mass transfer are analysed. Their environmental impacts and energy saving potential are illustrated in [3]. Camargo et al. [4] present a mathematical model for direct evaporative cooling air conditioning system, and its validation with experimental results [5]. Moreover, Franco et al. present a mathematical and experimental observation of implementation of evaporative cooling in greenhouses. Finally, Heidarinejad et al. [7] and Bozorgmehr et al. [8] present heat and mass transfer modelling of evaporative air coolers. The above mentioned models are based on work by Dowdy and Karabash [9] where correlation for determination of the convective heat transfer coefficient in a rigid cellulose evaporative medium is introduced. The correlation is based on assumption that air flow is turbulent and introduction of a characteristic length, which counts occupied volume inside the wetted medium or the pad.

Other authors proposed the correlation between heat transfer coefficient and input air velocity in the Direct Evaporative Pad (DEP) or the cooler. The correlation is established by using the measurement data of the DEP manufacturer. Correlation parameters depend on material properties and configuration of wetted medium used in the cooler [10], and models that implement this correlation are presented in [11, 12]. Rogdakis et al. [13] estimate the water temperature influence on the direct evaporative cooler operation with an additional correlation to determine heat transfer coefficient as function of mass air flow.

The above mentioned models treat the cooler as "black-box", where the outlet values are determined by using empirical correlations for definition of processes inside the device. This paper presents the numerical model, based on the physical background used for solving the air states throughout the cooler.

The motivation is to develop the model based on following propositions:

- Define the general physical model for heat and mass interaction on air/water interface;
- Define heat and mass transfer coefficients on the basis of the fluid flow theory;
- Establish connection with a thermodynamic solver or a database;
- Compare the results with measurement data in the particular example;
- Demonstrate the transfer coefficients variation throughout the cooler and
- Compute the output air states and water consumption.

This model should accurately compute the output air states and water consumption depending on different climate conditions. Such model presents the basis for an extensive model that is a tool for optimisation, cost savings and optimal control of the direct evaporative cooler placed in front of an air-cooled condenser of a refrigeration system.

Recently, new approaches for analyzing heat and mass transfer in laminar moist air flow inside vertical plate channels with falling water film are demonstrated [14]. Moreover, this has been shown for direct air cooling of forced convection [15], as well.

The general direct evaporative cooler is shown in Figure 1a. The basic elements are: wetted medium, water pump and air fan. Water circulates above the wetted medium where a small quantity evaporates in the air and the rest falls in the sump. The pump pushes water to the distribution above the pad, from where water pours onto the medium. We assume the use of a metallic DEP, designed by parallel metallic layers, as wetted medium. Since water is poured on the metallic layers from the top, the air is in direct contact with water. Water film is formed on the layers' surfaces, while air flows in horizontal direction between them. In this way air gets cooled and humidified at the same time. The schematic presentation of the physical model considered here is presented in Figure 1b.

The DEP effectiveness is defined as the ratio of the difference between input and output air temperatures and the difference between dry-bulb and wet-bulb input air temperatures. After all, water on wet-bulb temperature is the theoretical temperature to which air can be cooled inside the cooler. Here, subscripts a, v and w are denoted for air, water vapour and water, respectively¹. Therefore, the

¹Air temperature and humidity are denoted as T and w without a subscript

effectiveness is expressed by

$$\varepsilon = \frac{T_{in} - T_{out}}{T_{in} - T_{wb}}, [-], \quad (1)$$

where subscripts in and out represent input and output values.

In addition, the water used for air cooling evaporates in the cooler. The water consumption is proportional to air mass flow throughout the pad and to humidity difference between the outlet and the inlet air:

$$\Delta \dot{m}_w = \dot{m}_a (w_{out} - w_{in}), [\text{kg}_w/\text{h}]. \quad (2)$$

Absolute air humidity is defined as ratio between the mass vapour and the mass of dry air, such as:

$$w = \frac{m_v}{m_a}, \left[\frac{\text{kg}_v}{\text{kg}_a} \right]. \quad (3)$$

(a) Direct water-air evaporative pad

(b) Physical model considered in this paper

Figure 1: Schematic presentations of evaporative cooler and the physical model

In order to present our findings, this paper is organized as follows: the physical model is developed in the second section, validation of the developed model is presented in the third and results of the model are presented in the fourth part. At the end, conclusion gives the summery of the presented work.

2 Physical Model

Physical model used for this study is explained by defining the general equations for energy and mass balance for water/air interface interaction. For model to be closer to the realistic scenario, it accounts for

the internal fluid flow between parallel plates. To complete the picture, the physical model is connected to an external database.

2.1 Governing Equations

Here we explain governing principles of water/air interaction interface. The enthalpy of moist air is treated as the mixture of enthalpies of air and water vapour [18], such as:

$$h_m = h_a + h_v = c_{pa}T + w(c_{pv}T + h_{fg}) \quad (4)$$

where c_{pa} and c_{pv} are specific heats of air and vapour, respectively, and h_{fg} is the enthalpy of water vaporization.

If water is in direct contact with air in an open space, heat and mass transfers continuously occur. The steady-state energy equation of moist air with mass flow $\dot{m}_a$ over an infinitesimal water surface (dA), can be expressed as:

$$\dot{m}_a dh_m = \delta\dot{Q}_S + \delta\dot{Q}_L, \quad (5)$$

where $\delta\dot{Q}_S$ and $\delta\dot{Q}_L$ are denoted as sensible and latent heat on the air/water interface. The sensible heat is

$$\delta\dot{Q}_S = h_T(T_w - T)dA, \quad (6)$$

where the heat transfer coefficient is introduced as h_T , $[W/m^2K]$. Therefore, the driving force for heat transfer is the temperature difference between the water and the air. If the water temperature is lower than that of the air, the air is cooled, and vice versa. Further, the latent heat is defined with the water evaporation rate $d\dot{m}_w$ by

$$\delta\dot{Q}_L = d\dot{m}_w(c_{pv}T_w + h_{fg}). \quad (7)$$

Using heat/mass analogy, the water evaporation rate from the water surface to the air can also be expressed as:

$$d\dot{m}_w = h_M \rho_a (w_s - w)dA, \quad (8)$$

where w_s , h_M , $[m/s]$ and ρ_a represent the humidity of the saturated air film at water temperature, the mass transfer coefficient and the air density, respectively. Here, it is assumed that the water is covered by the saturated air film with the same temperature as the water. The driving force for mass transfer

is the difference between partial pressures of the saturated water vapour at temperature equal to water temperature and that of the water vapour in the air. If this difference is positive, the water evaporates, and if it is negative, the air vapour condensates. Also, it can be defined that the water evaporation rate is equal to the humidity change in air mass flow, such as

$$d\dot{m}_w = \dot{m}_a dw. \quad (9)$$

Three different cases can be observed in air cooling processes, where the water temperature is lower than the air ($T_w < T$). In the first case, the saturated air humidity is lower than the air humidity ($w_s < w$), the sensitive and latent heats go from the air to the water, both are negative, and air enthalpy reduces (cooling and condensation process). In other two cases, the saturated air humidity is higher than the air humidity ($w_s > w$), so the water evaporates into the air ($\dot{Q}_L > 0$). In the second case, the sum of the sensible and the latent energy is still negative ($\dot{Q}_S > \dot{Q}_L$), the air enthalpy as well as the air temperature decrease, but the air humidifies. In the third case, the sum of the sensible and the latent energy becomes positive ($\dot{Q}_S < \dot{Q}_L$), energy transfers from the water to the air, the air enthalpy grows, but, at the same time, the air temperature decreases and the humidity increases [16].

When the air streams over water surface parallel with x direction (x - the longitudinal coordinate), the infinitesimal water surface can be expressed as $dA = Pdx$, where P, [m] is width/perimeter of the water surface. If Eqs.(4,6,7,8,9) are substituted into in Eq.(5), one can obtain:

$$(c_{pa} + wc_{pv}) \frac{dT}{dx} = \left[\frac{Ph_T}{\dot{m}_a} + c_{pv} \frac{dw}{dx} \right] (T_w - T). \quad (10)$$

Also, by combining Eq.(8) and Eq.(9), we get

$$\dot{m}_a \frac{dw}{dx} = Ph_M \rho_a (w_s - w). \quad (11)$$

In the cooler, the water circulates in a closed cycle and only evaporated rate is added. Consequently the water inside is in the thermodynamic equilibrium with the input air. Therefore, the water temperature is on the wet-bulb temperature of the input air. Further, the humidity of the saturated air film on the water surface is determined assuming that the relative humidity is 100 % at the wet-bulb temperature. Hence,

$$T_w = T_{wb}(T_{in}, \varphi_{in}) \text{ and} \quad (12)$$

$$w_s = w(T_w, \varphi = 100\%) . \quad (13)$$

Since, the wet-bulb temperature is lower than the dry-bulb temperature, the sensible heat is negative. Additionally, the humidity of the saturated air film is higher than the air humidity, causing the water to evaporate into the air. Consequently, the air stream through the DEP belongs to the constant wet-bulb temperature line on the psychometric chart. The enthalpy of humid air slightly grows, like in the third case explained above.

Therefore, the governing equations that describe the air states valid on the air/water interface are Eqs.(10,11) with additional correlations expressed by Eqs.(12,13). For solving the governing equations with air/water interface conditions, the model needs to be connected with a psychometric function or a database for determination of the humid air properties.

The system of ordinary differential equations is solved for two profiles, temperature and air humidity

$$T = T(x) \text{ and } w = w(x). \quad (14)$$

Profiles alongside the pad need to be computed in the one-dimensional geometry. The length of domain is L , where L represents the water surface length.

Two thermodynamic values define the humid air state. The input air parameters are taken at the beginning of the DEP. The input (dry-bulb) temperature and humidity (T_{in} and w_{in}) can be determined by knowing the input state. The water temperature and the humidity of saturated air film are determined by Eq.(12) and Eq.(13), respectively.

2.2 Fluid Flow between Parallel Plates

Presented physical model takes into account both heat and mass transfer during moist air flows between parallel plates and water evaporation that occurs. Even though, the fluid momentum equation is not part of the presented physical model, the heat and mass transfers occur while the air flows between parallel plates. Because of that, the model uses analytical correlations developed for internal flows in order to determine heat and mass transfer coefficients. The analytical correlations are obtained from literature [17] and shortly reviewed here.

The Reynolds number for internal flow is:

$$Re_{D_h} = \frac{UD_h}{\nu_a} , \quad (15)$$

where U is the mean velocity, D_h is the hydraulic diameter and ν_a is the kinetic viscosity of air. For internal flows between parallel plates, the hydraulic diameter is

$$D_h = 2D, \quad (16)$$

where D is the distance between plates, as shown in Figure 1b. Reynolds number determines the fluid flow regime. The air flow between parallel plates is in the laminar regime, since Reynolds number is far below 2,000 in all observed examples.

The fluid flow begins with an entrance region, where velocity and temperature profiles change with the longitudinal direction. Later, the fluid flow moves to fully-developed region, for which boundary is determined by an entrance length. The boundary layer thickness for the laminar flow is defined by

$$\delta \sim 5x \sqrt{\frac{U_x}{\nu_a}}, \quad (17)$$

whereas the entrance length is calculated by:

$$X = 0.04 D Re_D; \quad Re_D = \frac{UD}{\nu_a}. \quad (18)$$

In the entrance region, the Nusselt number is determined by a dimensionless longitudinal coordinate, $x_* = (x/D_h)/(Re_D Pr)$, using the following correlation:

$$Nu(Sh) = \begin{cases} 1.849x_*^{-1/3} & x_* \leq 0.001 \\ 7.541 + 6.874(10^3 x_*)^{-0.488} e^{-245x_*} & x_* > 0.001. \end{cases} \quad (19)$$

The correlation is valid for uniform temperature of plates. Furthermore, Nusselt number between parallel plates, as function of the dimensionless longitudinal coordinate, is presented in Figure 2 [17]. The ordinate value is presented in the logarithmic scale, because Nusselt number rapidly decreases with the dimensionless longitudinal coordinate.

Apart of the heat transfer, the water evaporation occurs. The heat/mass transfer analogy is used, once again. The same correlation as given in Eq.(19) for determination of Sherwood number as the function of the longitudinal coordinate is used. For Sherwood number, the dimensionless longitudinal coordinate is $x_* = (x/D_h)/(Re_D Sc)$, where Prandtl is replaced by Schmidt number.

Prandtl and Schmidt numbers are defined as:

$$Pr = \frac{\nu_a}{\alpha_a}; \quad Sc = \frac{\nu_a}{D_{va}}, \quad (20)$$

where α_a represents the thermal diffusivity, $\alpha_a = k_a / \rho_a c_{pa}$, k_a is the air thermal conductivity and D_{va} is the diffusivity of vapour in air. The heat and mass transfer coefficients are expressed from Nusselt and Sherwood numbers as:

$$h_T(x) = \frac{k_a Nu}{D_h}; \quad h_M(x) = \frac{D_{va} Sh}{D_h}. \quad (21)$$

The heat and mass transfer coefficients depend on the longitudinal position between the parallel plates in the entrance region. Nusselt and Sherwood numbers are calculated by Eq.(19) and the heat and mass transfer coefficients by Eq.(21). The coefficients are used in the governing equations, Eqs.(10,11), where the air/water interface interaction is considered. Therefore, water flow inside the pad is ignored and it is assumed that air steams throughout the vertical water plates inside the pad.

Considering the laminar flow, the velocity profile between plates is unchanged in the fully-developed regime. Because of that, Nusselt and Sherwood numbers do not depend on the longitudinal location. Nusselt and Sherwood numbers [17] are in the fluid flow between parallel plates:

$$Nu = Sh = 7.54. \quad (22)$$

Consequently, the heat and mass transfer coefficients are constant in the fully-developed region.

Figure 2: Nusselt number in the entrance region between parallel plates

2.3 Thermodynamic and Thermophysical Properties

The presented physical model needs to be connected with a psychrometric solver or a database that calculates thermodynamic values of the moist air. The dry and wet bulb temperature, the absolute and relative humidity, the enthalpy, the dew temperature and the specific volume, as the thermodynamic properties of the humid air, are calculated by CoolProp solver [19]. The solver is based on equations expressed in ASHRAE [18]. Based on knowledge of two properties, the solver calculates other thermodynamic variables of the humid air state at the atmospheric pressure.

The specific heat, the conductivity and the viscosity of moist air are calculated, using the same source [19]. The diffusion coefficient of vapour in air - D_{va} , ($[m^2/s]$) is treated as temperature dependent. Due to the fact that the process happens at the atmospheric pressure, the equation used in the model is taken from the literature [20]:

$$D_{va} = 1.87 \cdot 10^{-10} T^{2.072}. \quad (23)$$

3 Model Validation

The analytical correlations for determination of heat and mass transfer coefficients in the specific geometry are described in Subsection 2.2. For calculation of coefficients in a particular example, it is necessarily to know the hydraulic diameter, the length of plates, the input air velocity and the pad size. The developed model with given analytical correlations inside parallel plates is validated for Oxyvap DEP² [21].

This section starts with the description of the Oxyvap DEP used for validation. Then we explain the implemented model based on the developed physical model. The numerical model calibration is used to specify accuracy of the results. In order to minimize the error between the numerical results and measurement data, the values of the hydraulic diameter is optimised. Lastly the capabilities of the developed numerical model are demonstrated by comparing results with the measurement data for various climate conditions.

3.1 Direct Evaporative Cooler

The Oxyvap cooler is a highly efficient DEP, placed inside an aluminium frame with integrated water distribution system. It is used in a wide variety of applications that requiring cooling and/or humidi-

²The Oxyvap - the direct evaporative cooler is designed by the OXYCOM company

fication. The design is presented in Figure 3. The pad is manufactured to have longitudinal, vertical layers with the distance of 4.16 mm between them (D_f). Layers are produced from aluminium sheet of thickness 0.70 mm (δ_f). The DEP thickness is 90.00 mm. The air flows between layers that consist of 2.00 mm longitudinal fins (L_f). Therefore, the number of fins in the air stream is 45. The density area (β) is $549 \text{ m}^2/\text{m}^3$ and the effective contact area is 1.141 times higher than if the layers were consisted of simple plates. The DEP parameters are given on the right hand side of Figure 3.

Figure 3: The Oxyvap cooler - the DEP used for model validation

3.2 Description of Implemented Model

The governing equations (Eq.(10,11)) describe the heat and mass transfer on the air/water interface, in general. By using Eqs.(19,20,21) for the heat and mass transfer coefficients, the model concentrates on the air flow between parallel plates covered by water films. So, the model can be used for solving the air temperature and humidity longitudinal profiles in between the parallel fins covered by falling water. The presented metallic shape inside the DEP (Figure 3) is very complex and the air streams between fins subsequently.

The idea is to use the developed model sequentially, so the output data from previous computation becomes inputs for the next. The number of consecutive computations is equal to the number of fins alongside the air stream. The number of fins that air streams throughout is 45 ($L/L_f = 45$). Accordingly, the outlet temperature and humidity of the air between the parallel fins become the input values for the

next fins pair. The thermodynamic and thermophysical properties are recalculated every time before the next computation. The input air velocity before following parallel fins is determined in accordance with the air mass flow balance.

3.3 Numerical Calibration

The discretisation of the governing equations is performed by the classical second-order finite difference numerical scheme. The code is developed in MATLAB software. The numerical calibration results are presented in Table 1. This example uses the input parameters: $T_{in} = 35^\circ\text{C}$, $\varphi_{in} = 35\%$ and $v_{in} = 2.5\text{ m/s}$. The hydraulic diameter is selected to be 4.0 mm. In the next subsection, determination of the hydraulic diameter as the free parameter in the model is demonstrated. The DEP effectiveness, CPU time and errors between numerical computations for various number of nodes over the parallel fins for discretisation are presented. The total number of nodes for discretisation of governing equations over parallel fins is defined as N . Resulting error for numerical calibration is introduced by

$$\zeta_{N/\bar{N}} = \frac{|\varepsilon_N - \varepsilon_{\bar{N}}|}{\varepsilon_{\bar{N}}}, [-], \quad (24)$$

where $\bar{N}$ represents the reference number of nodes.

Table 1: DEP effectiveness, CPU time and errors between numerical results ([%]) obtained by different number of nodes for discretisation

$N, [-]$	$\varepsilon, [\%]$	time, [s]	$\zeta_{N/100}$	$\zeta_{N/500}$	$\zeta_{N/1000}$	$\zeta_{N/5000}$	$\zeta_{N/10000}$
50	86.28	1.0	0.1927	0.4258	0.4717	0.5235	0.5335
100	86.45	1.0	-	0.2234	0.2796	0.3314	0.3414
500	86.65	1.5	-	-	0.0462	0.0981	0.1081
1,000	86.69	4.3	-	-	-	0.0520	0.0620
5,000	86.73	312.2	-	-	-	-	0.0100
10,000	86.74	2397.2	-	-	-	-	-

With increasing the number of nodes for the discretisation of the governing equations, the resulting error becomes smaller. If the result with $\bar{N} = 10,000$ is used as the referent, the error is less than 1 % by using 50 nodes for discretisation over parallel fins. In the example with 5,000 nodes, this error is 0.01 %.

If the criteria for accuracy of numerical results is selected to be less than 0.1 %, the numbers of nodes that satisfy this criteria are $N = 1,000$ and higher. In following computations, it is decided that the number of nodes for the discretisation of the governing equations over the pair of fins is 1,000. The CPU time with the selected number of nodes for discretisation is about 4 s computed on MacBook Pro, 2.7

GHz Intel Core i7 and 16 GB RAM.

3.4 Determination of Effective Hydraulic Diameter

The presented DEP has very complex geometry. The finned aluminium sheet or layer is designed for maximum evaporation and small pressure drop in the very small thickness. The heat and mass transfer coefficients are dependent on Reynolds number, Eq.(15), which is a function of the particular hydraulic diameter. In this model, Reynolds number is defined as

$$Re_{\tilde{D}_h} = \frac{v_{in} \tilde{D}_h}{\nu_a}, \quad (25)$$

where v_{in} is input velocity before parallel fins and $\tilde{D}_h$ is the effective or particular hydraulic diameter.

The effective hydraulic diameter for the model is determined by the numerical experiments. Set of numerical computations are performed and their results are compared with the available measurement data. The measurements were done by the Netherlands institute, TNO. The comparison is performed for outlet temperature in various climate conditions and input air velocities. Temperature and humidity are varied in intervals from 20 to 50 °C and from 10 to 80 %, respectively, as well as the input velocity is 1, 2, or 3 m/s. The available number of measurement data is 280. The discrepancy between the model and experimental results is quantified by:

$$\zeta_{ijk} = \frac{|T_{out}^m - T_{out}^e|}{T_{in} - T_{out}^e} \Big|_{ijk}, \quad (26)$$

where subscripts i,j,k represent different output values for diverse inputs: temperature, humidity and input velocity. The superscripts m and e represent model and experimental results, respectively. The criteria for determining the optimal effective hydraulic diameter is the minimum of sum of discrepancies in all available points for comparison. The sum is calculated by:

$$\Delta = \sum_{ijk} \zeta_{ijk}. \quad (27)$$

Sums of discrepancies for different particular hydraulic diameters are presented in Table 2. The effective hydraulic diameter is varied from 3.00 mm to 4.00 mm. The optimal effective hydraulic diameter for the developed numerical model is calculated to be 3.40 mm. In this example, the sum of discrepancies is 0.4223, the lowest observed. Regarding input conditions, the maximum output temperature discrepancy of 3.68 % is observed for input parameters: $T_{in} = 38^\circ\text{C}$ and $\varphi = 10\%$ and $v_{in} = 2\text{ m/s}$.

The particular distance between parallel layers/fins is two times smaller than the effective hydraulic diameter, Eq.(16). Therefore, the model is based on the assumption that horizontal air flows between parallel water films with distance between them 1.70 mm ($\tilde{D} = 1.70$ mm).

Table 2: Numerical computation of optimal hydraulic diameter in the model

$\tilde{D}_h$, [mm]	3.00	3.20	3,40	3.60	3.80	4.00
Δ , [-]	0.7096	0.5285	0.4223	0.4336	0.5027	0.5746

The effective hydraulic diameter ($\tilde{D}_h$) is smaller than the distance between layers/fins inside the DEP (D_f). We can conclude that, there are following reasons. The aluminium layer has the thickness of 0.70 mm and layers are covered by water films. The water film thickness is dependent from water Reynolds number, as defined in literature [22]. In observed example, implementing the Nusselt theory of water film thickness, the average thickness of water film is cca 0.32 mm. Yu et al. [23] confirms that water film thickness in this water regime could be up between 0.28 and 0.38 mm. Of course, the water film narrows free area for air streams. Because of that, the free distance for air movement between the parallel fins is maximum 2.80 mm. Therefore, the compact heat exchanger philosophy have to be implemented for quantified interference between subsequent fins on the air stream [24].

3.5 Comparison between Numerical Results and Measurement Data

The outlet temperatures are used to compare numerical results and the measurement data. The measurement data are available in the spectra of dry-bulb temperatures (from 20°C to 50°C) and relative humidities (from 10% to 80%) and for three input velocities 1, 2 and 3 m/s. The results computed by the numerical model, measurements and their relative difference for few examples of input temperatures and relative humidities and the input air velocity of 1 m/s are presented in Table 3³. The agreement between the model and experimental results is very good, with the maximum observed difference of less than 2.00 %. The similar demonstration of the model accuracy can be presented for input velocities 2 and 3 m/s, where the available measurement data exist.

Average relative differences for three velocities, 1, 2 and 3 m/s are 0.55, 1.34 and 0.33 %, respectively. It can be concluded, that with previously determined hydraulic diameter, the numerical model gives very good results in comparison with the measurement data. Therefore, the numerical model is validated and

³n/a - non available data

it can be used for further analysis.

Table 3: Comparison of outlet temperatures computed by the developed numerical model and measurement data available ($v_{in} = 1 \text{ m/s}$ and $\bar{D}_h = 3.40 \text{ mm}$).

		$\varphi_{in}, [\%]$								
		30			50			70		
		T_{out}^e	T_{out}^m	$\zeta, [\%]$	T_{out}^e	T_{out}^m	$\zeta, [\%]$	T_{out}^e	T_{out}^m	$\zeta, [\%]$
$T_{in}, [^\circ\text{C}]$	48	31.2	31.14	0.20	n/a	37.20	n/a	n/a	42.08	n/a
	44	28.1	28.23	-0.46	n/a	33.82	n/a	n/a	38.40	n/a
	40	25.3	25.34	-0.18	30.5	30.47	0.10	n/a	34.72	n/a
	36	22.5	22.48	0.10	27.1	27.13	-0.11	n/a	31.06	n/a
	32	19.8	19.62	0.89	23.8	23.81	-0.04	27.4	27.41	-0.03
	28	17.0	16.77	1.33	20.6	20.50	0.47	23.8	23.77	0.13
	24	14.1	13.92	1.28	17.3	17.21	0.54	20.2	20.14	0.31
	20	11.1	11.05	0.47	14.0	13.91	0.61	16.6	16.52	0.51

4 Model Results

In this section, results obtained by the developed numerical model are presented. In the results, we give the heat transfer and mass transfer coefficients inside the DEP and influences of climate data and the air velocity on the temperature and humidity profiles.

4.1 Heat and Mass Transfer Coefficients

The air flow inside the pad is laminar, because the highest Reynolds number is 642 (Eq.(15)) for selected hydraulic diameter 3.40 mm and the maximum input air velocity of 3 m/s. However, the length of the fin inside the DEP is 2 mm only. The boundary layer thickness at the end of the longitudinal fin (Eq.(17) and the entrance length (Eq.(18) for three input air velocities are given in Table 4.

Table 4: The entrance length and the thickness of boundary layer at the end of longitudinal fin in variation of input air velocities

v_{in}	[m/s]	1	2	3
δ_{Lf}	[mm]	0.89	0.63	0.52
X	[mm]	7.28	14.55	21.82

Even with the input air velocity of 1 m/s, the entrance length is about three times longer than the length of the fins. The thickness of boundary layer with this velocity is on the order-of-magnitude of the half of the distance between layers/fins. Therefore, the air flow throughout the parallel water films is in the entrance region and the transfer coefficients vary with the longitudinal coordinate, as described in Subsection 2.2. On the other hand, Nusselt and Sherwood numbers are constant in the fully-developed

region (Eq.(22)) and the heat and mass transfer coefficients in this example are $h_T = 59 \text{ W/m}^2\text{K}$ and $h_M = 0.0603 \text{ m/s}$, respectively.

The transfer coefficients are presented for input air properties: $T_{in} = 34^\circ\text{C}$ and $\varphi_{in} = 40\%$. The input air velocity is chosen to be 2 m/s . Variation of the mass transfer coefficients for air flow throughout the DEP is presented in Figure 4. Both transfer coefficients decrease rapidly across the longitudinal fin. However, air moves over the fins in a row. Therefore, the coefficients at the beginning of the subsequent fin are much higher. In this way, the average value of the transfer coefficients is kept high, as in the case with a compact heat exchanger. The entrance length ($X = 14.55 \text{ mm}$) is bigger than the length of the fins ($L_f = 2.00 \text{ mm}$), so the air flows along the longitudinal fins in the entrance region. For better understanding, the heat transfer coefficient across the first five fins in the DEP is shown in Figure 5. Here, the heat transfer coefficient varies from $1,011 \text{ W/m}^2\text{K}$ at the beginning of the fin to $83 \text{ W/m}^2\text{K}$ at its end. The average heat transfer coefficient is $117 \text{ W/m}^2\text{K}$, two times higher than the heat transfer coefficient in the fully-developed region between long parallel plates assuming the same particular distance between them. As one can see, the presented DEP belongs to compact heat exchanger and heat and mass transfer coefficients are high.

At the beginning of the fin, the heat and mass transfer coefficients are not defined by Eq.(19). An assumption that the first two discretisation nodes on the fins have the second derivative equal is applied to solve this singularity at the boundary of the fin.

During air stream throughout the pad, the density increases and the air velocity drops. The air velocity before the last fin inside the pad is 1.95 m/s , instead of starting 2.00 m/s . This is the reason why transfer coefficients slightly decline along the cooler length. The heat transfer coefficient across the last, 45th fin is between 993 and $82 \text{ W/m}^2\text{K}$.

The additional analysis of influence of the length of the fins (L_f) on the heat and mass transfer coefficients is given in Table 5. The same pad thickness and particular hydraulic diameter are used here, but the number of the longitudinal fins (N_f) varies. If the cooler is consisted of vertical layers without any fins, the fin length is the same as the pad thickness. The average heat and mass transfer coefficients, in this case, are: $\bar{h}_T = 61 \text{ W/m}^2\text{K}$ and $\bar{h}_M = 0.0620 \text{ m/s}$, respectively. Therefore, the DEP effectiveness is 71.8% . With increasing the number of fins over layers, the transfer coefficients grow and the DEP

Figure 4: The mass transfer coefficient profile used in numerical computation

Figure 5: The heat transfer coefficient profile across the first fifth fins on the layer

Table 5: Variation of the heat and mass transfer coefficients and the DEP effectiveness with variation of the number of fins on layers in the same pad thickness

N_f	$[-]$	1	10	30	45	90
L_f	$[mm]$	90.0	9.0	3.0	2.0	1.0
h_T	$[W/m^2K]$	60	77	104	117	145
h_M	$[m/s]$	0.0604	0.0743	0.0984	0.1109	0.1371
ε	$[%]$	71.8	80.3	88.7	91.5	95.3

effectiveness gets larger. The maximum of the average heat and mass transfer coefficients are observed in the example of the fin length of 1 mm. The DEP effectiveness in this example is more than 95 %. Also, three cases of the heat transfer coefficients are shown in Figure 6 with the number of fins 1, 10 and existing 45. The graphical presentation shows that the heat transfer coefficient inside the presented cooler ($N_f = 45$) is constantly higher than in the case of simple parallel layers without any fin ($N_f = 1$).

Figure 6: Heat transfer coefficients throughout the pad with different number of longitudinal fins

4.2 Temperature and Humidity Profiles

Temperature and humidity profiles throughout the DEP with three selected input velocities are presented in Figure 7. The input air parameters are: $T_{in} = 34^\circ C$ and $\varphi_{in} = 40\%$. Passing throughout the pad, air temperature declines and humidity rises. Inside the pad, the water temperature is assumed to be

equal to the wet-bulb temperature for input air properties (Eq.(12)). In presented example, the wet-bulb temperature is 23.16 °C. In Figure 7, decreasing profiles trends are shown. This happens due to decrease in the temperature difference between the water and the air as air passes deeper inside the cooler, where the air becomes more humid.

Furthermore, some waves are obvious on the presented profiles. The waves are consequence of the heat and mass transfer coefficients profiles across the fins, as presented in Figures 4 and 5. With increase the heat and mass transfer coefficients at the beginning of the fin, temperature and humidity vary faster and as the air flows to the end of the fin, values vary slower. In the end of the fin, air inflows between the subsequent pair of fins and the heat and mass transfer coefficients, again, become higher, making the values vary faster.

The DEP effectiveness depends on the input air velocity. By increasing the velocity, the effectiveness reduces. The influence of air velocity is twofold. The transfer coefficients (Eq.(19)) via Reynolds number (Eq.(15)) rise as the air velocity grows, while the mass air flow in governing equations (Eqs.(10,11)) is proportional to the air velocity. So, the DEP effectiveness varies from 84.8 to 98.2 % and the output humidity increases from 0.0040 to 0.0045 kg_v/kg_a.

The second example shows influence of the relative humidity of the input air on the temperature and humidity profiles. The input temperature and the velocity are fixed and equal to 30 °C and 2 m/s. The cases with relative humidities of 10, 40 and 70 % are given in Figure 8. The relative humidity influences the wet-bulb temperature of the air, while lower humidity enables higher driving forces for the heat and mass transfer on the air/water interface.

The last example varies input (dry-bulb) air temperature. Three selected cases are presented in Figure 9 with input temperatures 15, 25 and 35 °C. The relative humidity profiles are almost equal here. Similar is seen with temperature profiles, but with different initial condition. The examples are presented for the input velocity of 2 m/s and the relative humidity of 30 %.

Summary of presented results is given in Table 6. In the first example, with the air velocity of 1 m/s, the DEP effectiveness is more than 98 %. In the second example, the DEP effectiveness is almost not influenced by changing the relative humidity of input air. But, for the water consumption is the opposite situation. The water consumption is much higher with declines of the relative humidity of input air.

Figure 7: Temperature and humidity with variation of input air velocity. The input air properties are: $T_{\text{in}} = 34^{\circ}\text{C}$ and $\varphi_{\text{in}} = 40\%$

Figure 8: Temperature and relative humidity with variation of the input relative humidity. Examples are given for the input air temperature $T_{\text{in}} = 30^{\circ}\text{C}$ and the input air velocity $v = 2 \text{ m/s}$

Figure 9: Temperature and relative humidity with variation of input air temperature. 30 % is the input relative humidity and 2 m/s is the input air velocity

The third example shows slightly varying of DEP effectiveness with grows of the input air temperature.

Furthermore, the water consumption is almost proportional with temperature increases.

Table 6: Output results of presented examples

$T_{in} = 34\text{ }^{\circ}\text{C}$, $\varphi = 40\%$ (Figure 7)				
v_{in}	T_{out}	ε	w_{out}	Δw
[m/s]	[$^{\circ}\text{C}$]	[%]	[kg _v /kg _a]	[kg _v /kg _a]
1	23.35	98.2	0.0179	0.0045
2	24.08	91.5	0.0177	0.0043
3	24.83	84.5	0.0174	0.0040
$T_{in} = 30\text{ }^{\circ}\text{C}$, $v_{in} = 2\text{ m/s}$ (Figure 8)				
φ_{in}	T_{out}	ε	w_{out}	Δw
[%]	[$^{\circ}\text{C}$]	[%]	[kg _v /kg _a]	[kg _v /kg _a]
10	14.72	91.0	0.0091	0.0064
40	20.93	91.2	0.0145	0.0039
70	25.89	91.4	0.0207	0.0018
$\varphi = 30\%$, $v_{in} = 2\text{ m/s}$ Figure 9				
T_{in}	T_{out}	ε	w_{out}	Δw
[$^{\circ}\text{C}$]	[$^{\circ}\text{C}$]	[%]	[kg _v /kg _a]	[kg _v /kg _a]
35	22.66	91.5	0.0159	0.0053
25	15.39	90.8	0.0100	0.0041
15	8.01	89.9	0.0061	0.0029

5 Conclusion

The numerical model for solving heat and mass transfer in a metallic-compact direct evaporative cooler is presented in this paper. The physical model is developed using the energy equation of humid air and the mass balance on the air/water interface. The model is capable of calculating the humid air properties inside the cooler. The analytical correlations for heat and mass transfer coefficients for air flowing between parallel plates are incorporated in this model. The model is based on assumption that air flows between parallel water walls which are on wet-bulb temperature of the input air. In the particular example, the particular hydraulic diameter is determined by numerical experiments. Numerical results given by the model fit accurately to the available measurement data. The high effectiveness is obtained with the evaporative pad of 90 mm thickness. The air velocity determines the pad effectiveness, lower velocities gives higher effectiveness, and vice versa. The relative humidity drastically influences on the water evaporation rate, and no influence on effectiveness. The input dry-bulb air temperature grows slightly increases the effectiveness and almost linearly the water consumption. The developed model can be used as the tool for optimization of refrigeration systems, where this direct evaporative pad is used as pre-cooler for air-cooled condensers and other applications.

Acknowledgments

The first author want to thank Government of Montenegro for National Scholarship for Excellence for research stay at KU Leuven, Department of Mechanical Engineering, Division of Applied Mechanics and Energy Conversion.

Authors want to thank OXYCOM company for the measurement data of Oxyvap pad used for the model validation.

References

- [1] World Bank Technical Paper no. 421, Energy Series, Evaporative Air-Conditioning, Applications for Environmentally Friendly Cooling, The International Bank for Reconstruction and Development, The World Bank, 1999.
- [2] Y. M. Xuan, F. Xiao, X. F. Niu, X. Huang, S. W. Wang, Research and application of evaporative cooling in China: A Review (I) - Research, Renewable and Sustainable Energy Reviews 16, 3535-3546, 2012.
- [3] Y. M. Xuan, F. Xiao, X. F. Niu, X. Huang, S. W. Wang, Research and application of evaporative cooling in China: A Review (II) - System and equipment, Renewable and Sustainable Energy Reviews 16, 3523-3534, 2012.
- [4] J. R. Camargo, C. D. Ebinuma, S. Cardoso, A mathematical model for direct evaporative cooling air conditioning system, Engenharia Térmica 4, 30-34, 2003.
- [5] J. R. Camargo, C. D. Ebinuma, J. L. Silveira, Experimental performance of a direct evaporative cooler operating during summer in a Brazilian city, International Journal of Refrigeration 28, 1124-1132, 2005.
- [6] A. Franco, D. L. Valera, A. Peña, Energy efficiency in greenhouse evaporative cooling techniques: cooling boxes versus cellulose pads, Energies 7, 1427-1447, 2014.
- [7] G. Heidarinejad, M. Bozorgmehr, Heat and mass transfer modeling of two stage indirect/direct evaporative air coolers, ASHARE Journal - Thailand Chapter, 1-8, 2007-2008.
- [8] R. D. Deshmukh, S. Jdeshmukh, D. A. Warke, Theoretical analysis on heat and mass transfer in a direct evaporative cooler, International Journal of Innovative and Emerging Research in Engineering, 2/1, MEPCON, 2015.
- [9] J. A. Dowdy, N. S. Karabash, Experimental determination of heat and mass transfer coefficients in rigid impregnated cellulose evaporative Media, ASHRAE Transactions 93/2, 382-395, 1987

- [10] J. M. Wu, X. Huang, H. Zhang, Theoretical analysis on heat and mass transfer in a direct evaporative cooler, *Applied Thermal Engineering* 29, 980-984, 2009
- [11] J. M. Wu, X. Huang, H. Zhang, Numerical investigation on the heat and mass transfer in a direct evaporative cooler, *Applied Thermal Engineering* 29, 195-201, 2009
- [12] A. Fouda, Z. Melikyan, A simplified model for analysis of heat and mass transfer in a direct evaporative cooler, *Applied Thermal Engineering* 31, 932-936, 2011
- [13] E. D. Rogdakis, I. P. Koronaki, D. N. Tertipis, Estimation of the water temperature Influence on direct evaporative cooler operation, *International Journal of Thermodynamics* 16/4, 172-178, 2013.
- [14] C. Ren, Y. Wan, A new approach to the analysis of heat and mass transfer characteristics for laminar air flow inside vertical plate channels with falling water film evaporation, *International Journal of Heat and Mass Transfer*, 103, 1017-1028, 2016.
- [15] V. I. Terekhov, M. V. Gorbachev, H. Q. Khafaji, Evaporative cooling of air in an adiabatic channel with partially wetted zones, *Thermophysics and Aeromechanics*, 23/2, 221-230, 2016.
- [16] W. F. Stoecker, J. W. Jones, *Refrigeration and Air Conditioning*, McGraw Hill, 1982.
- [17] A. Bejan. *Convective Heat Transfer*, 4th edition, Wiley, 2013.
- [18] ASHRAE, *American Society of Heating, Refrigeration and Air Conditioning Handbook - Fundamentals*, ASHRAE, 2001.
- [19] I. H. Bell, J. Wronski, S. Quoilin, V. Lemort, Pure and Pseudo-pure Fluid Thermophysical Property Evaluation and the Open-Source Thermophysical Property Library CoolProp, *Industrial & Engineering Chemistry Research* 53, 6, 2498-2508, 2014.
- [20] Y. A. Çengel, A. J. Ghajar, *Heat and Mass Transfer, Fundamentals & Application*, 5th Edition, McGraw Hill Education, 2015.
- [21] Oxy-com company. <http://www.oxy-com.com/C1235-Our-Company.html>, Technical Documentation, 2016.

- [22] D. W. Zhou, T. Gambaryan-Roisman, P. Stephan, Measurement of water falling film thickness to flat plate using confocal chromatic sensing technique, *Experimental Thermal and Fluid Science* 33, 273-283, 2009.
- [23] Y. Q. Yu, S. J. Wei, Y. H. Yang, X. Cheng, Experimental study of water film falling and spreading on a large vertical plate, *Progress in Nuclear Energy* 54, 22-28, 2012.
- [24] R. M. Manglik, A. E. Bergles, Heat transfer and pressure drop correlations for the rectangular offset strip fin compact heat exchanger, *Experimental Thermal and Fluid Science* 10, 171-180, 1995.
- [25] L. E. Sissom, D. R. Pitts, *Elements of Transport Phenomena*, McGraw-Hill Inc, USA, 1972.
- [26] F. P. Incropera, D. P. DeWitt, T. L. Bergman, A. S. Lavine, *Fundamentals of Heat and Mass Transfer*, 6th edition, John Wiley & Sons, USA, 2007.

Highlights

Revised manuscript ATE-2016-13898

THE NUMERICAL MODEL FOR DIRECT EVAPORATIVE COOLER

- The numerical model for metallic-compact cross-flow air/water direct evaporative cooler is developed;
- The effective hydraulic diameter is determined by a parameter estimation using experimental data available from a producer of the direct evaporative pad;
- Temperature and humidity of air throughout the cooler and pad effectiveness can be simulated by the developed numerical model;