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NUMERICAL ANALYSIS OF HEAT AND MASS TRANSFER PROCESSES THROUGH THE MAISOTSENKO CYCLE

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ABSTRACT

This report describes numerical modelling of heat and mass transfer in exchangers utilizing the Maisotsenko Cycle for indirect evaporative cooling. For this purpose, numerical models are developed based on the modified ε-NTU method to perform thermal calculations of the indirect evaporative cooling process, thus quantifying overall performance of considered heat exchangers. Numerical simulation reveals many unique features of considered HMX (heat and mass exchanger), enabling an accurate prediction of its performance. The results of computer simulation showed high efficiency gains that are sensitive to various inlet conditions, and allow for estimation of optimum operating conditions, including suitable climatic zones for the proposed unit using.

NOMENCLATURE

c_p	[J/(kg·K)]	Specific heat capacity of moist air
F	[m^2]	Surface
h	[m]	Height
G	[kg/s]	Moist air mass flow rate
i	[kJ/kg]	Specific enthalpy of the moist air
L, l	[m]	Streamwise length of cooler
M	[kg/s]	Water vapor mass transfer rate
NTU	[-]	Number of transfer units, $\mathrm{NTU} = \alpha F/(Gc_p)$
$\hat{Q}$	[kW/m^3]	Specific cooling capacity rate respected to 1 m^3 of heat exchanger's filling
s	[m]	Fin pitch
t	[°C]	Temperature
v	[m/s]	Velocity
W	[W/K]	Heat capacity rate of the fluid
x	[kg/kg]	Humidity ratio
X	[m]	Coordinate along primary air flow direction
Y	[m]	Coordinate perpendicular to X coordinate
Z	[m]	Coordinate along fins direction

Special characters:

α	[W/(m^{2}K)]	Convective heat transfer coefficient
β	[kg/(m^{2}s)]	Mass transfer coefficient
δ	[m]	Thickness
λ	[W/(m·K)]	Thermal conductivity
η	[%]	Effectiveness
σ	[-]	surface wettability factor, $\sigma \in (0.0 \div 1.0)$

Non dimensional coordinates:

Le	[-]	Lewis factor $\mathrm{Le} = \alpha/(\beta c_p)$
NTU	[-]	Number of transfer units $\mathrm{NTU} = \alpha F/(Gc_p)$
$\bar{X}$	[-]	$\bar{X}=X/l$ – relative X coordinate
$\bar{Y}$	[-]	$\bar{Y}=Y/l$ – relative Y coordinate
$\bar{Z}$	[-]	$\bar{Z}=Z/h_{fin}$ – relative Z coordinate

Subscripts

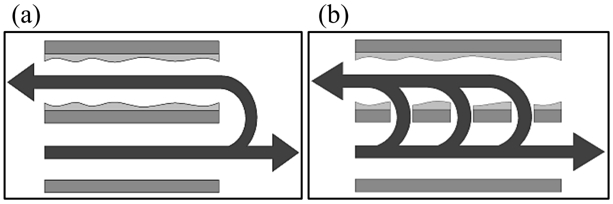
1	Main (primary) air flow
2	Working (secondary) air flow
cond	Heat transfer by thermal conduction
Fin	Fins
g	Water vapor
i	Inlet
l	Latent heat flow
o	Output
p	Plate surface
plt	Channel plate
product	Referenced to the product part of the heat exchange
s	Sensible heat flow
w	Water
WB	Wet bulb temperature
work	Referenced to the working part of the heat exchange
X	Air streamwise in the dry channel
Y	Air streamwise in the wet channel
●	Referenced to the elementary plate surface
●●	Referenced to the elementary fin surface
′	Condition at the air–plate interface temperature
′′	Respected to the plate surface

INTRODUCTION

The building sector accounts for a major part of the world's total end energy consumption. It has the largest single potential for improving the efficiency of energy use. Cooling energy is an important part of this energy and the demand for cooling is continuously increasing due to the growing demand for better indoor comfort conditions in buildings. The use of water evaporation for decreasing air temperature is a well-known cooling technology and an environmental friendly application.

Due to increase in the awareness of environmental problems resulting from green house gas emissions, various applications of evaporative cooling have been extensively studied and used for industrial and residential sectors, such as: humidifier, cooling tower and evaporative cooler [1 and 2]. Direct and indirect evaporative cooling systems are used to produce low temperature medium fluid (i.e., water, air, etc.). Direct evaporative cooling system adds moisture to the cool air, which also makes conditions more uncomfortable for humans as (air) humidity increases. On the other hand, an indirect evaporative cooling system provides only sensible cooling to the process air without any moisture addition. In an indirect evaporative air cooler, water is separated from the primary (product) air using the heat exchanging plate [3]. During operation, evaporation of the water occurs in one side of the plate where the secondary (working) air moves along; while the primary (product) air flows across the other side. Evaporation of the water causes reduction of the temperature of the plate, resulting in heat transfer between the primary air and the plate. Therefore, it is more attractive than direct evaporative system. However, conventional indirect evaporative cooling has a serious thermodynamic limitation: the ultimate temperature of the processes is the ambient air wet bulb temperature. In this regard, our study focuses on the analysis of heat and mass transfer process in the novel evaporative air cooler based on Maisotsenko cycle (M-Cycle) [3–5,9–13], which allows to realize sub-wet bulb evaporative cooling without using vapor compression machine. The main idea of the novel heat and mass exchanger (HMX) is manipulating the air flow by splitting the working air from the part of main air flow, which is indirectly pre-cooled before it enters the wet passage. The first time the M-Cycle technology was proven was in 1984. Currently the Coolerado Corporation produces several air conditioners (commercial, residential, solar and hybrid) relying only on the M–Cycle. The National Renewable Energy Lab (NREL) from USA tested the Coolerado's air conditioners documenting that they are up to 80% more efficient than traditional systems [5]. There are many types of HMXes which are realizing the M-Cycle flow scheme (cross-flow, regenerative flow, regenerative flow with perforations and other- Figure 1).

(a) (b)



(c) (d)

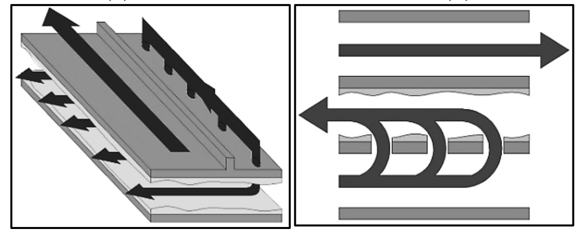


Figure 1 Different versions of M-cycle heat and mass exchangers: (a) regenerative HMX; (b) regenerative HMX with perforation; (c) cross-flow HMX; (d) modified regenerative HMX

The M-Cycle uses the same wet side and dry side of a plate as the conventional indirect evaporative air cooler but with a significantly different airflow configuration wherein a main stream of air is passed along a dry channel, simultaneously passing an auxiliary air stream counter currently along a moist channel which is exchanging heat with the airflow in the dry passage (Fig. 1). It gives opportunity to get cold air below the wet bulb temperature and approaching nearly dew point temperature of the ambient air in a theoretically optimal situation.

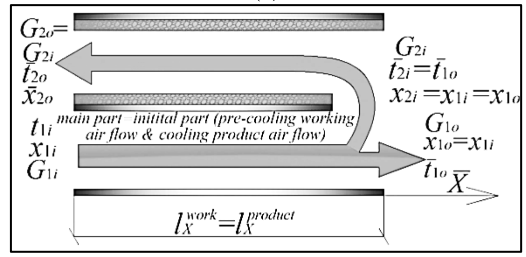
With increasing interest in renewable energy, it becomes important to study the most effective exchangers for evaporative air cooling. That is why authors performed detail theoretical study comparing different types of the M-Cycle HMXes. One of the best methods of analysing indirect evaporative air coolers performance is a multi-module computer simulation, based on a mathematical model. In this paper five different exchangers with the M-Cycle will be put to comparative analysis (Fig. 1a–c):

-Regenerative HMX (variant 1- V1)
-Cross-flow HMX (variant 2- V2)
-Perforated regenerative HMX with 5 holes (variant 3- V3)
-Perforated regenerative HMX with 10 holes (variant 4- V4)
-Perforated regenerative HMX with 20 holes (variant 5- V5)

METHODS

In many problems, including considered one, an engineer is not interested in receiving thermodynamic variables distributions normal to the heat exchanger plate surface, but bulk average values. Therefore we gave preference to ε–NTU–model. In this case air stream in matrix passages is considered as a gaseous fluid flow with constant temperature, velocity and mass transfer potential (humidity ratio of air) in the direction normal to plate surfaces, which is equal to bulk average values [6–9].

(a)



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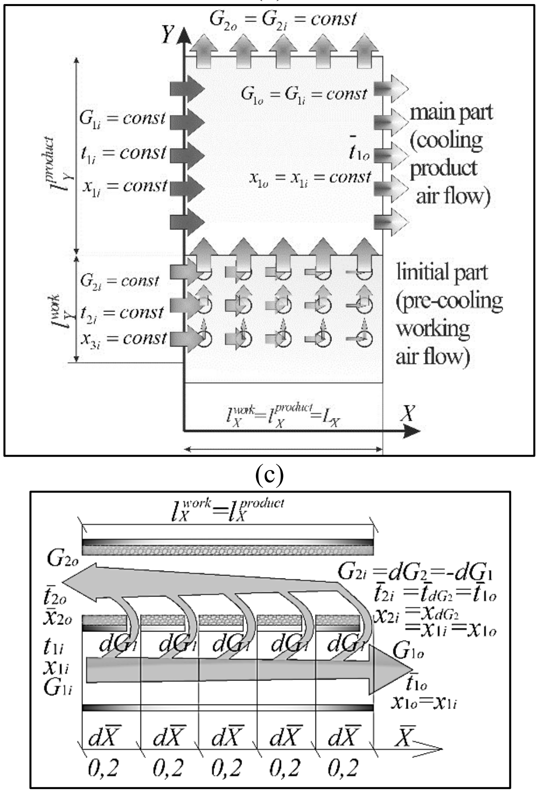


(c)

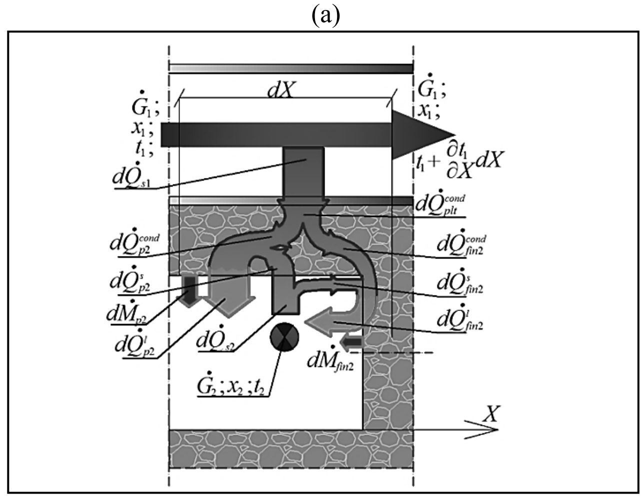
Figure 2 Assumptions: (a) regenerative HMX (V1); (b) cross-flow HMX (V2); (c) regenerative HMX with 5 holes (V3);

Authors have developed the original ε–NTU–model describing coupled heat and mass transfer in considered the M-Cycle heat exchangers. Proposed mathematical models were based on the following assumptions:

- Steady state operation;
- Heat losses to the surroundings are negligible;
- Airflow is an ideal, incompressible gas mixture of dry air and water vapors;
- Kinetic properties of air flow and water film are constant and equal to bulk average values;
- Driving force of mass transfer is a gradient of moisture content (vapor's partial pressure);
- Longitudinal molecular diffusion of water vapor in air and longitudinal heat conduction along the wall as well as inside the fluids in the direction of air flow are negligible;
- Consumed water rate corresponds to sufficient evaporation and keeping up the material of plates in hygroscopic saturated condition. This causes that air flow heat capacity is much bigger than water's– $W \gg W_w$
- Exchangers' filling is made of porous material, which allows for even distribution of water in channels;
- Wet and dry channels are separated by impenetrable metal foil;
- In HMXes V1 and V3-V5 secondary air flow and primary air flow are flowing through the same dry channels, therefore secondary air is pre-cooled before entering wet channel (Fig. 2(a) and (c));
- In cross-flow the M-Cycle HMX secondary air stream flows at firs in the dry, initial channel which is separated from the main part. At the end of initial dry channel all air flow enters wet part of the HMX (Fig. 2(b));
- Perforation in initial part of the cross-flow HMX is even and very dense;
- In HMXes V3-V5, there is finite number of holes (5, 10 and 20) in the exchanger's plate along the *X*–axis in each cross-section (ex. HMX V3 $\bar{X} = X/l_X = 0{,}2;\ 0{,}4\ldots$ etc. – Fig. 2(c)). In each cross-section of the *X*–axis, 1/5,1/10 or 1/20 of the total secondary flow gets to the wet channel (regardless of the number of holes along the *Y*– axis) [11]. A small number of perforations was adopted for a detail analysis of the impact of perforation density on cooling effectiveness.

Mathematical model describing cross-flow the M-Cycle HMX (V2) will be presented in this paper. Models for other HMXes are analogous (model for regenerative HMX was published in [12] and model for perforated regenerative HMX was published in [11]). The mathematical model of the cross flow type the M-Cycle evaporative cooler was developed on the heat and mass balance considerations in the form of partial differential equations made up for separated air streams and realized in orthogonal coordinate system

(a)



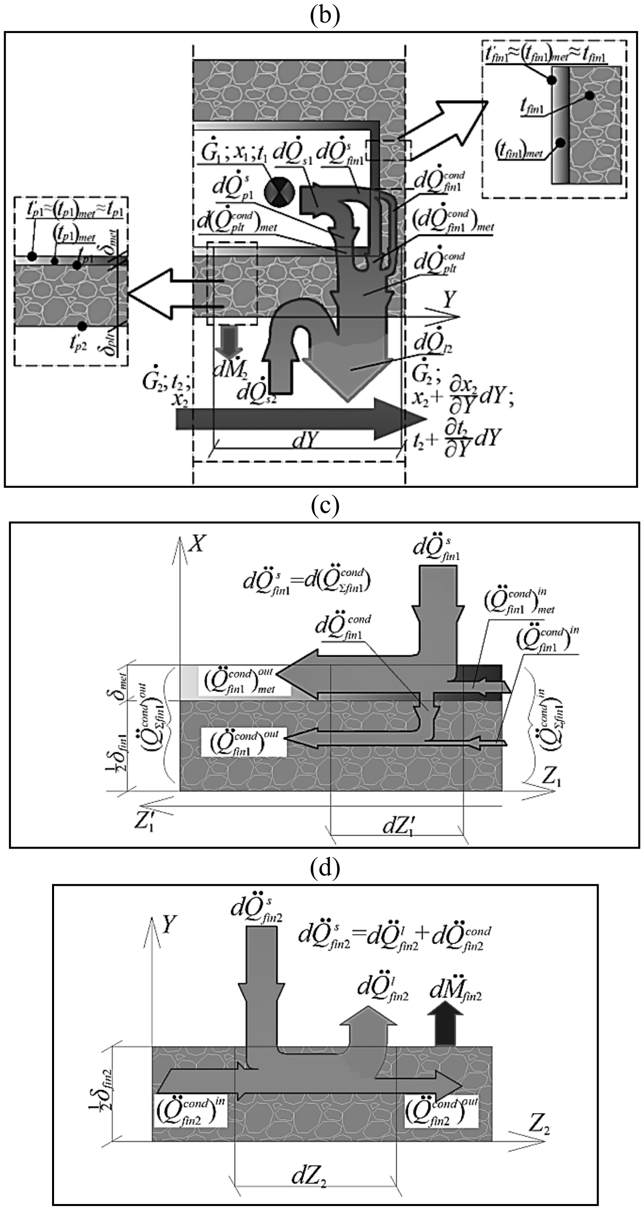


Figure 3 Schematic of heat and mass transfer in differential control volumes: (a) detail view on the wet channel; (b) detail view on the dry channel; (c) for the dry channel fin; (d) for the wet channel fin

The following balance equations can be written for the air streams passing through elemental area in the product part of the exchanger (see Fig. 3 (a) and (b))

- For the air stream in the dry channel:
  - ➢ The energy balance:

$$\dot{G}_1 c_{p1}\frac{\partial t_1}{\partial X}dX = -d\dot{Q}_1^s = -\left(d\dot{Q}_{p1}^s + d\dot{Q}_{fin1}^s\right) \tag{1}$$

where $\dot{G}_1 = \left(dY/L_Y\right)G_1$

- For the air stream in the wet channel:
  - ➢ The energy balance:

$$\begin{aligned}\dot{G}_2 c_{p2}\frac{\partial t_2}{\partial Y}dY &= -d\dot{Q}_2^s + \left[c_{g2}\left(t'_{p2}-t_2\right)d\dot{M}_{p2} +\right.\\ &\left.+c_{g2}\left(\bar{t}_{fin2}-t_2\right)d\dot{M}_{fin2}\right] = -\left[d\dot{Q}_{p2}^s + d\dot{Q}_{fin2}^s\right] +\\ &+\left[c_{g2}\left(t'_{p2}-t_2\right)d\dot{M}_{p2} + c_{g2}\left(\bar{t}_{fin2}-t_2\right)d\dot{M}_{fin2}\right]\end{aligned} \tag{2}$$

- ➢ The mass balance for the water vapor inside the wet channel

$$\dot{G}_2\frac{\partial x_2}{\partial Y}dY = d\dot{M}_2 = \left(d\dot{M}_{p2} + d\dot{M}_{fin2}\right) \tag{3}$$

where $\dot{G}_2 = \left(dX/L_X\right)G_2$.

The energy balance for a differential control volume of the fin in the dry channel can be presented as the heat conduction equation considering only sensible heat transfer on the surface of the fin (see Fig. 3 (c))

$$\begin{cases}\left(\ddot{Q}_{\Sigma fin1}^{cond}\right)^{in} + d\ddot{Q}_{fin1}^{s} = \left(\ddot{Q}_{\Sigma fin1}^{cond}\right)^{out}\\ \left(\ddot{Q}_{\Sigma fin1}^{cond}\right)^{in} = \left(\ddot{Q}_{fin1}^{cond}\right)^{in} + \left(\ddot{Q}_{fin1}^{cond}\right)_{met}^{in}\\ \left(\ddot{Q}_{\Sigma fin1}^{cond}\right)^{out} = \left(\ddot{Q}_{fin1}^{cond}\right)^{out} + \left(\ddot{Q}_{fin1}^{cond}\right)_{met}^{out}\end{cases} \tag{4}$$

Rearranging the set (4) and assuming, that $t'_{fin1} = \left(t_{fin1}\right)_{met} = t_{fin1}$, the heat conduction equation for the composite fin can be written as

$$\frac{\partial^2 t_{fin1}}{\partial \bar{Z}_1^2} = -m_{fin1}^2 h_{fin1}^2\left(t_1 - t_{fin1}\right) \tag{5}$$

where:

$$m_{fin1}^2 = \frac{2\alpha_1}{\lambda_{fin1}\delta_{fin1} + 2\left(\lambda_{fin1}\right)_{met}\left(\delta_{fin1}\right)_{met}} \tag{6}$$

Under the following boundary conditions:

$$t_{fin1}\Big|_{\bar{Z}_1=0} = t'_{p1} = t_{p1}\ ;\quad \partial t_{fin1}/d\bar{Z}_1\Big|_{\bar{Z}_1=1.0} = 0 \tag{7}$$

the differential equation (5) can be solved analytically

$$\left(\frac{\partial t_{fin1}}{d\bar{Z}_1}\right)_{\bar{Z}_1=0} = m_{fin1}h_{fin1}\left(t_1 - t_{p1}\right)th\left(m_{fin1}h_{fin1}\right) \tag{8}$$

$$\left(t_{fin1} - t_1\right) = \left(t_{p1} - t_1\right)\frac{ch\left[m_{fin1}h_{fin1}\left(1-\bar{Z}_1\right)\right]}{ch\left(m_{fin1}h_{fin1}\right)} \tag{9}$$

The energy balance for a differential control volume of the fin in the wet channel should be described taking into account combined sensible and latent heat exchange on the surface of the fin (Fig. 3(c)):

$$\left(\ddot{Q}_{fin2}^{cond}\right)^{in} + d\ddot{Q}_{fin2}^{s} = d\ddot{Q}_{fin2}^{l} + \left(\ddot{Q}_{fin2}^{cond}\right)^{out} \tag{10}$$

Additionally the M-cycle mathematical model was supplemented by energy balance equations developed

- For the plate surface in the dry passage (see Fig. 3 (a),(b))

$$d\left(\dot{Q}_{plt}^{cond}\right)_{met} = d\dot{Q}_{p1}^{s} \tag{11}$$

which can be converted taking into account the following set of energy balance equation

$$\begin{cases} d\dot{Q}_{plt}^{cond} = d\left(\dot{Q}_{plt}^{cond}\right)_{met} + d\dot{Q}_{\Sigma fin1}^{cond} \\ d\dot{Q}_{\Sigma fin1}^{cond} = d\dot{Q}_{fin1}^{s} \end{cases}; \tag{12}$$

to the following form

$$\dot{Q}_{plt}^{cond} = d\dot{Q}_{p1}^{s} + d\dot{Q}_{fin1}^{s} \tag{13}$$

- For the filling surface
- At the wet passage (see Fig. 3(a)–(b))

$$d\dot{Q}_{1}^{s} + d\dot{Q}_{2}^{s} = d\dot{Q}_{2}^{l} \tag{14}$$

which can be converted to the form

$$\left(d\dot{Q}_{p1}^{s} + d\dot{Q}_{fin1}^{s}\right) + d\dot{Q}_{p2}^{s} = d\dot{Q}_{p2}^{l} + d\dot{Q}_{fin2}^{cond} \tag{15}$$

that significantly simplifies iterative procedure of the local plate temperature computation at each node of integration step.

To derive the unique decision, the set of simultaneous partial differential equations (1),(2),(3), (10), (13), (15) is supplemented by the boundary conditions, establishing initial thermodynamic parameters values of exchanged air streams on the entries to appropriate channels of the product part of the heat exchanger [10–12].

Furthermore, integration algorithm of the governing differential equations is supplemented with additional algorithm describing mixing process of the air stream in the wet channel with air stream from the dry channel. This algorithm is used for calculation of differential equations for V2 HMX [10] and perforated regenerative HMX (V3-V5) [11].

VALIDATION OF NUMERICAL MODEL FOR CROSS-FLOW M-CYCLE HMX (V2)

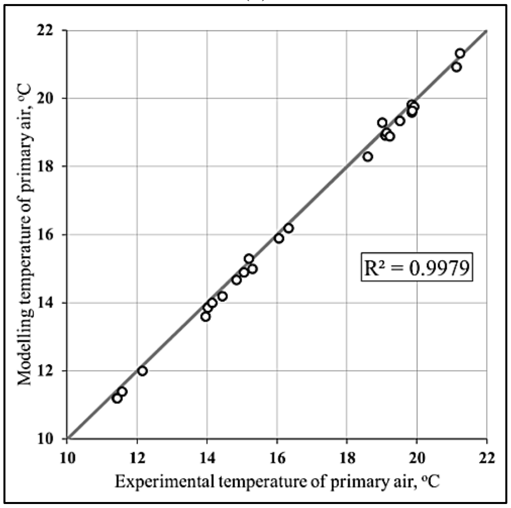
The cooling performance of the cross-flow the M-Cycle evaporative cooler predicted by simulation was validated against experimental data collected by Weerts [13] on the testing bench at The National Snow and Ice Data Center, USA . The model was set to the same operating conditions as for experimental cases: the same inlet air flow temperature and relative humidity and the same air flow rates. This analysis established the accuracy of the model in predicting the performance of the real indirect evaporative air cooler with Maisotsenko cycle.

Weerts' studies were performed for inlet air flow values varying between 1095 m^3/h to 4047 m^3/h (when supply to working air ratio is equal 1:1), air entrance temperature varying between 17.3°C (63.1 °F) and 32.3°C (90.1 °F) and air entrance %RH varying from 7.4% to 62.9%. Weerts measured inlet and outlet parameters of primary and working air streams. For the product and working air temperature and relative humidity measurement, HMP45C sensors (manufacturer: Campbell Scientific) with 0.05°C and 2.0% RH measure accuracy were used. For inlet air temperature and %RH, ZW-007 (manufacturer: HOBO) sensors, with 0.21°C and 2.5% RH measure accuracy were used. A hot wire anemometer 407123 (manufacturer: Extech Instruments) was used to determine the velocity profile for air flow, with 3.0% accuracy. All these measurement sensors were linked to a CR10X data logger (manufacturer: Campbell Scientific) and a computer for data recording and analyses. The signals were scanned and reported at 3 minute intervals. The data was also saved at the same intervals as well.

Modeling results compared with experimental data collected by Weerts, including the output primary and working air temperature and working air humidity ratio are shown respectively in figures 4 (a)–(c). Simulations were performed, varying the intake air temperature and relative humidity within the accepted range, while keeping the other parameters unchanged. Deviation of the primary stream outlet temperature is up to 5%, the coefficient of determination between the experimental and the simulation data is equal 0.9979 (Fig. 4(a)). For the working air flow, the correlation is for output temperature is a little lower and it's equal 0.9813 (Fig. 4(b)). For the secondary air stream output humidity ratio, coefficient of determination is equal 0.9711; maximal discrepancies are equal 0.6 g/kg (Fig. 4(c)).

Equations describing regenerative HMXes models are analogous to ones describing cross-flow HMX, therefore its accuracy is considered to be similar.

(a)



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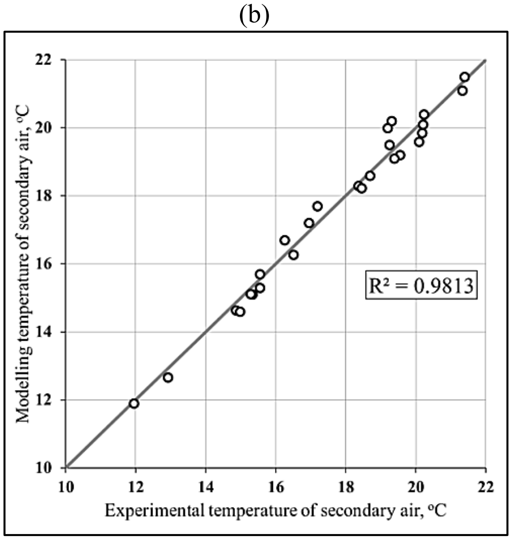

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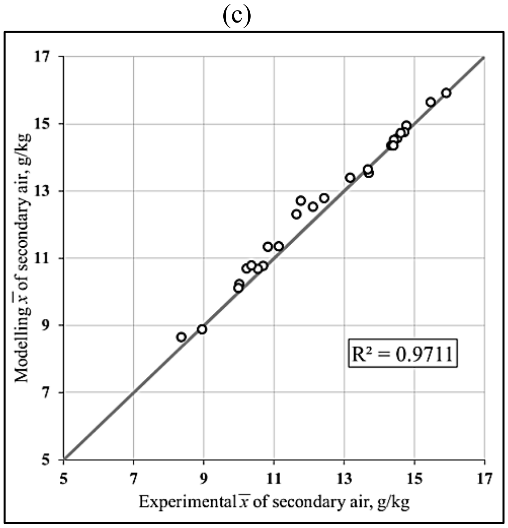


Figure 4. The M-Cycle mathematical model validation, comparing to Weerts data [13]: (a) output temperature- primary stream t_{1o}; (b) output temperature- secondary stream $\bar{t}_{2o}$; (c) output $\bar{x}_{2o}$- secondary stream.

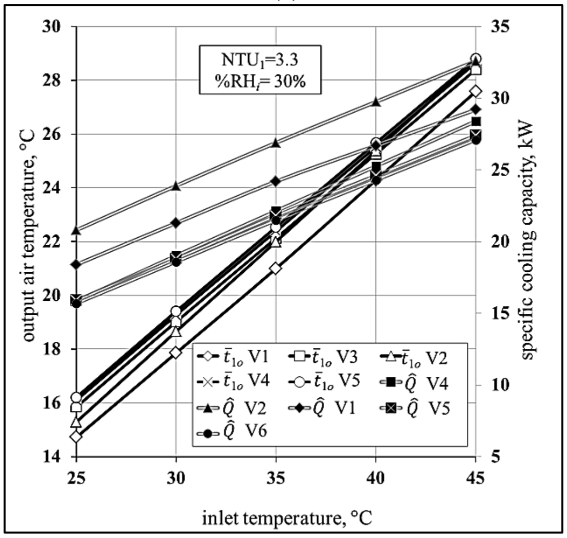
TRENDS AND RESULTS

Influence of inlet air flow parameters on cooling effectiveness

Varying the inlet air temperature between 25°C and 45°C while all other parameters remain unchanged, the simulation was carried out using the computer models. The results are presented in Figure 5(a). A trend in increasing supply air temperature and cooling capacity of all HMXes coincides with increasing inlet air temperatures. This is due to the fact that while secondary flow has constant inlet relative humidity, its mass transfer potential is almost constant, however, higher temperature facilitates the evaporation of the water (large temperature difference between the water film and the air), and therefore hotter air cools more effectively. It is noteworthy that relatively small temperature difference occurs between obtained inlet air output temperatures. Although the temperature of inlet air changes by 20 degrees (from 25°C to 45°C), the maximal difference in outlet temperatures is equal: V1: 12.9°C, V2: 13.5°C, V3, V4, V5: 12.6°C. Specific cooling capacity rate $\hat{Q}$ varies between 18.4–29.2 (V1), 20.8–32.6 kW (V2), 16.3–27.8 kW/m^3 (V3), 15.9–27.4 kW/m^3 (V4), 15.7–27.1 kW/m^3 (V5). This shows that this indirect evaporative HMX is more efficient at higher inlet air temperatures, suggesting it is more suited to a high-temperature environment.

Another set of simulations was performed keeping other parameters unchanged and varying relative humidity of inlet air. Results can be seen in Fig. 5(b)-(c). When the relative humidity of inlet air increased from 30 to 70% (for t_{1i}=30°C), accordingly the wet bulb effectiveness increased by 61% (V5) and 66% (V1), while outlet temperature has raised and the specific cooling capacity rate has dropped: respectively from 6.4 (V5) to 7.4°C (V1) and from 11.0 (V5) to 14.8 (V2) kW/m^3 respectively. Based on the lowest predicted output temperature value, the evaporative air cooler is most efficient in dry and hot climatic conditions (Fig. 5(a,b)), but the wet bulb temperature effectiveness is higher in the wet regions (Fig. 5(c)). For example: for the inlet air relative humidity RH_i=70% (t_{1i}=30°C) the temperature drop across considered HMXes was equal 4.3°C (V5) to 4.7°C (V1), which corresponds to the lowest value of this parameter in considered range of inlet air conditions, so showing the evaporative cooler to be redundant in a humid climate, but the wet bulb effectiveness for this inlet air conditions are at the highest level (Fig. 5(c)). Such paradoxical conclusion proves, that efficiency related to wet bulb or dew point temperature is not an adequate indicator for evaporative coolers performance, when it’s considered as the only efficiency factor.

(a)



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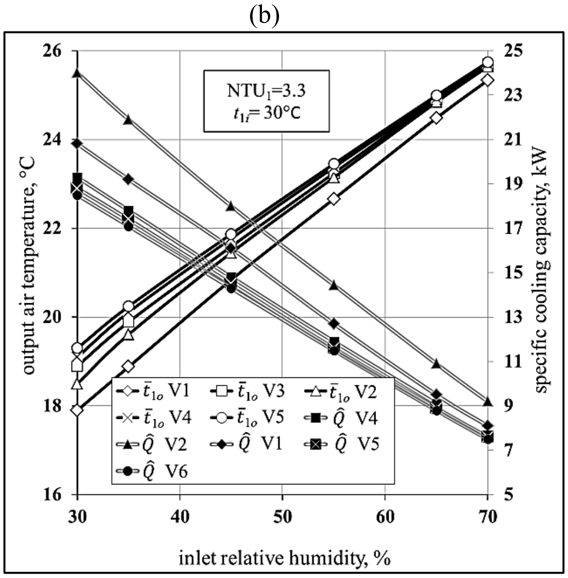

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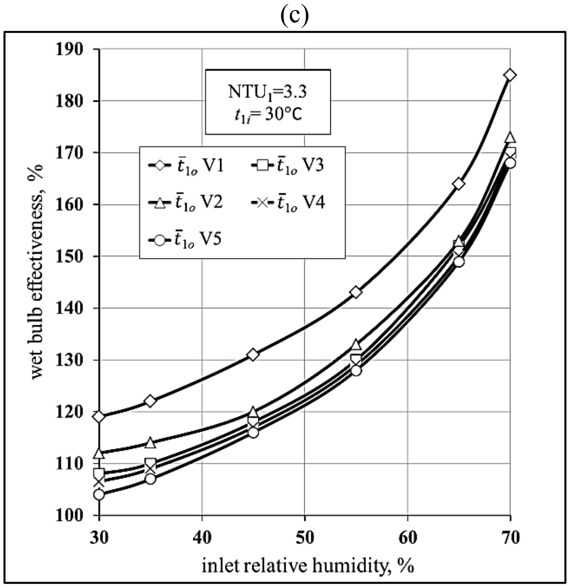

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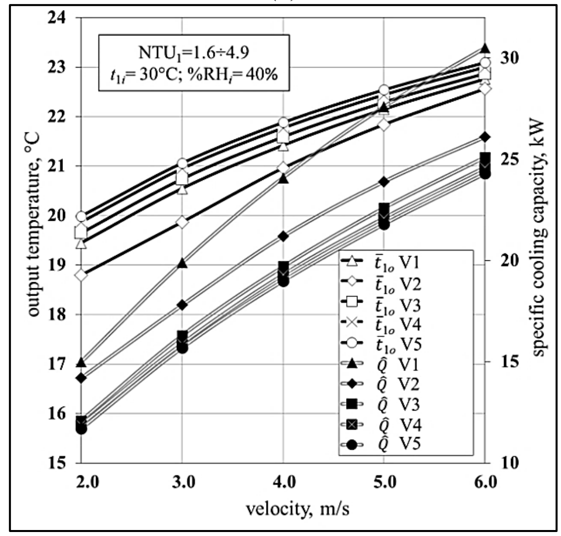

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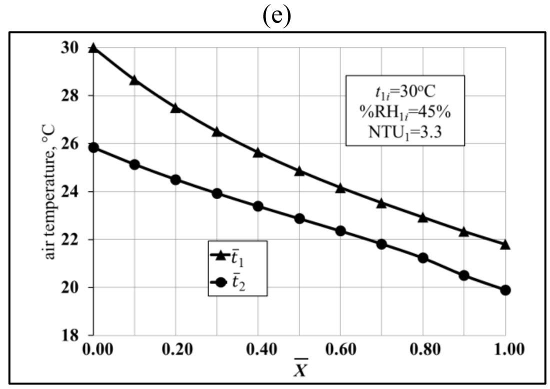

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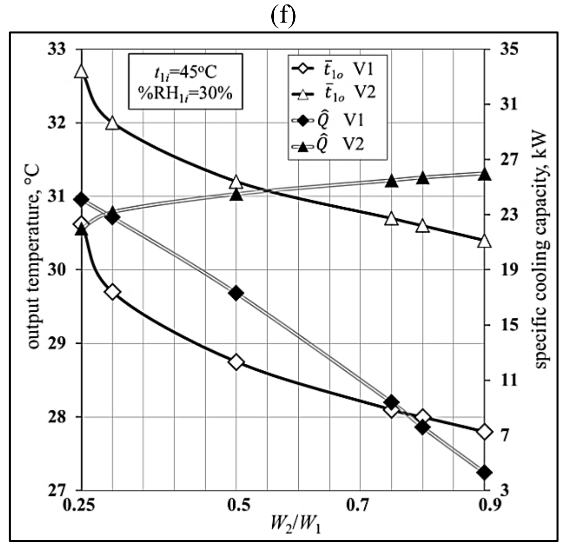

Figure 5 Numerical simulation results: (a) output temperature and specific cooling capacity- air flow with constant inlet temperature and variable relative humidity; (b) $\bar{t}_{1o}$ and $\hat{Q}$ air flow with constant inlet relative humidity and variable temperature; (c) $\bar{t}_{1o}$ and $\hat{Q}$- HMX V1 as function of secondary to primary air ratio; (d) average temperature profiles along $\bar{X}$ axis in product part of HMX V2; (e) wet bulb effectiveness for all HMXes- - air flow with constant inlet temperature and variable relative humidity

Influence of air flow velocity on cooling effectiveness

Simulations were carried out to investigate the effect of air flow velocity on the performance of considered HMXes. When declared primary air flow velocity increases (at fixed secondary to primary air mass flow rate ratio), the product air flow rate will increase in proportion, so does the working air flow rate in wet channels. Simulations were carried out, varying the air flow velocity from 2 to 6 m/s while keeping other parameters unchanged. The results of simulation are presented in Fig. 5(d). As shown in Fig. 5(d) the specific cooling capacity rate increases as air speed is increased (i.e with increasing air mass flow rate). However, at the same time, product air flow output temperature has raised: respectively from 18.8 to 22.6°C (V1), 19.5–22.8°C (V2), 19.7–22.9°C (V3), 19.9–23.0°C (V4) and 20.0–23.1°C (V5). Lower air stream velocity results in higher NTU value, which improves effectiveness of cooling, resulting in lower output temperature. Higher value of cooling capacity, when air stream velocity increases, follows from the fact, that air flow rate is higher. The drop in product air flow output temperature is relatively small, comparing to the change of air flow rate value, therefore overall cooling efficiency is higher in proportion with increasing air stream velocity. However, higher air speed in channels causes higher pressure drops. Often, limiting factor for the air cooler is the possibility to achieve lower output temperatures, because cooling capacity can always be increased by using several devices. That is why HMXes construction should guarantee relatively low value of air flow speed in the channels. However, air stream velocity shouldn't be too low, because then exchanger dimensions will became too large, to achieve any reasonable cooling capacity.

Comparison between considered exchangers

Multivariate simulations shown, that efficiency of considered HMXes is very similar: maximal differences in outlet temperatures were up to 1.5°C. This shows that other factors must be taken under consideration (for example hydraulic and economic) to fully compare considered devices. In all of presented studies (Figures 5(a) –(e)), lowest outlet temperature was achieved by regenerative HMX (V1), while highest specific cooling capacity was achieved by cross-flow exchanger (V2). Lowest values of cooling capacity and highest outlet temperatures were achieved by perforated regenerative HMX with the densest perforation (V5). It important to observe, that cross-flow exchanger shows higher temperature effectiveness than counter-flow exchangers V3-V5 and very similar temperature effectiveness to exchanger V1. This is due to the uniqueness of the M-Cycle. The initial part of the cross-flow HMX (Fig. 2b) pre-cools the secondary air flow before entering the main part of the exchangers. Secondary air temperature value is highest at the beginning of the exchanger and lowest at the end of the HMX. Temperature distribution in the wet channel and dry product channels created that way is then similar to temperature distribution in counter-flow exchangers (Fig. 5(e)): hot product air contacts with hotter working air, and cold product air contacts with cooler working air. That is why the M-Cycle cross-flow HMX effectiveness is significantly higher than typical indirect evaporative air coolers.

Another thing which is important to mention, is the fact that despite achieving lower temperature values, V1 HMX is always characterized by lower specific cooling capacity than V2 HMX (Fig. 5(a),(b),(d) and (f)). This follows from the fact, that construction of regenerative air cooler assumes, that both secondary and primary air stream are flowing through the same dry channel and at the end of the HMX they are separated (secondary air enters the wet channel, while product air is delivered to the conditioned apartments), while in cross-flow HMX the streams are separated before entering each channels. For example, at constant secondary to primary air ratio W_2/W_1=0.5, assuming inlet air flow rate in the dry channel equal 1000 m^3/h, outlet primary air flow for cross-flow HMX will be equal 1000 m^3/h, while for regenerative HMX only 500 m^3/h, because half of the air flow was returned to the wet channel. For secondary to primary air ratio equal 1, in regenerative HMXes, whole air flow is returned to wet channel and cooling capacity is equal 0. Different trends in cooling capacity for V1 and V2 HMX can be seen if Figure 5(f).

Although the V1 HMX shown higher temperature effectiveness, pure counter flow in the plate heat exchanger is very difficult to realize due to the geometry of the channels (plates) with air entering and leaving on the same sides. In this regard, it can be assumed that cross-flow the M-Cycle HMX is most reasonable for commercial purposes. Its temperature effectiveness is close to regenerative HMX, while its cooling capacity is higher and construction is easier to develop [3].

Impact of number of holes on cooling efficiency of the HMXes V3-V5

Multivariate numerical studies have shown that regenerative recuperator with smaller number of holes is characterized by higher temperature efficiency, and higher cooling capacity than its denser perforated equivalent (Fig. 5(a)–(d)). The example difference between output temperature of the product air flow amounted to considered units are (for input parameters t_{1i}= 30°C, %RH_{1i}= 45%):

- 21,6 °C for V3 HMX
- 21,8 °C for V4 HMX
- 21,9 °C for V5 HMX

It is easy to notice that cooling efficiency of heat exchangers is similar, so the meaningful comparison should take into account more factors. Lower cooling capacity of units with denser perforation is due to the fact, that warm air is entering wet channel through the entire length of the HMX. Positive effect of reducing secondary air stream humidity ratio which improves water evaporation (it is the result of wet and dry air streams mixing in the working air channel) was smaller in comparison with the negative influence of the rise of working air temperature (also caused by mixing process in the wet channel).

The higher number of holes increases the number of mixing points, so that the negative effect of heating the secondary air stream increases. Therefore the cooling capacity of the dense perforation recuperator is lower than the corresponding unit with a smaller number of holes. Dense perforation however generates lower pressure loss, and therefore to develop the optimum number of holes in considered recuperators it is necessary to carry out the compromise optimization method taking into account other performance indicators.

Different situation occurs when cross-flow HMX is taken under consideration. V2 HMX uses the cross-flow diagram of the main and secondary air, thus keeping dense perforations in in the dry working channel along secondary air flow (along X axis- Fig. 2(b)) direction does not adversely affect the efficiency of heat and mass transfer processes. It is important to analyze the number of perforated holes in the Y–axis direction (Fig. 2(b)). Too high density of holes can adversely affect the cooling efficiency of the recuperator, while too little can cause excessively high pressure loss. Further numerical analysis will allow determining the optimal solution.

CONCLUSIONS

This study presents theoretical energy analyses of the novel plate-fin heat exchangers based on the Maisotsenko cycle and used for indirect evaporative cooling in air conditioning systems.

Modified ε–NTU–model for the novel dew point indirect evaporative coolers based on accurate assumptions and mathematical analysis of the heat and mass transfer process inside M-cycle finned heat exchangers was proposed.

The performance of considered HMXes was investigated and parametrically evaluated by transitional simulation under various ambient and working/operating conditions in terms of cooling efficiency. The computation model results were validated against available experimental data. The positive results of this validation indicated that the sufficient accuracy in simulation could be obtained.

Sensitivity analysis was performed to determine which design parameters have the most impact on the HMX performance under various ambient and operating conditions in terms of dew point effectiveness and specific cooling capacity rate. From this analysis it was concluded, that:

- M-cycle technique is effective way of indirect evaporative cooling in hot and dry climates.
- the heat and mass transfer performance strongly depends on inlet air temperature and humidity, geometrical size of the channels, intake air velocity and secondary to primary air mass flow ratio
- wet bulb effectiveness η_{WB} (and dew point effectiveness) isn't an adequate indicator for indirect evaporative air cooler performance, when it's considered as only efficiency factor. In some cases (e.g. high inlet air relative humidity) wet bulb effectiveness can increase while specific cooling capacity rate of the HMX becomes lower. An optimized design of the M-cycle heat exchangers is a compromise between wet bulb effectiveness and cooling capacity.
- The temperature effectiveness of the cross-flow M-cycle HMX is similar to the counter-flow M-cycle HMXes, while its specific cooling capacity is higher and construction is easiest to design, therefore cross-flow the M-Cycle HMX seems to be the most reasonable unit for commercial purposes.
- Dense perforation has negative impact on cooling efficiency of regenerative HMX.

The results presented in this paper indicate the attractiveness of the novel M-Cycle heat exchangers and allow extending the potential of useful utilization of evaporative cooling for the purpose of air conditioning. The carried out analyses are expected to be beneficial to researchers and should be of great value in designing of indirect evaporative cooling units used for air conditioning systems.

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