Numerical modelling of a counter-flow regenerative evaporative cooler based on modified effectiveness-NTU method

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Abstract. This study develops a numerical model for a counter-flow regenerative evaporative cooler in Engineering Equation Solver environment by using modified effectiveness-NTU method. The overall performance of the counter-flow heat exchanger was predicted by varying inlet air temperature and humidity.

1. Introduction
Regenerative evaporative cooler, as the IECs with closed-loop configuration, has the ability to provide supply air at a temperature below the wet-bulb towards the dew-point temperature of inlet air theoretically [1]. According to the estimates [2], a counter-flow REC could reduce 13-58% of the total electrical power annually consumed by conventional compression air conditioners for various climates in China. Predominantly, there are two types of regenerative evaporative coolers including single-stage counter-flow and multi-stage M-cycle cross-flow configurations. More recently, there have been growing interests in developing the single-stage counter-flow regenerative evaporative coolers by numerical methods [3-5]. However, the effectiveness-NTU method hasn’t been developed in the modelling heat and mass exchanger. To fill the gap, this paper developed a dedicated numerical computer model for the proposed counter-flow heat and mass exchanger based on modified effectiveness-NTU method. The model was validated with published experimental data. We used the model to simulate the effect of inlet air temperature and humidity on the performance of counter-flow heat and mass exchanger.

2. Numerical modelling of the heat and mass exchanger

2.1 Physical model of the heat and mass exchanger
Fig. 1 shows the schematic of geometry and the operating conditions for the counter-flow plate heat and mass exchanger. In the dry channels of the heat exchanger, product air at $P_{pi}=0.1 \text{ MPa}$, $t_{pi}=38.9 ^\circ C$, $t_{wb,pi}=21.11$ enters one set of channels. The total volumetric flow rate of air passing through all of the dry channels is $V_p=2737 \text{ m}^3/\text{h}$. Portions of product air, as working air enters the other set of channels (wet channels) at $P_{wi}=P_{po}$ and $t_{wi}=t_{po}$ through the perforations at the end of dry channels. $R_{ratio}$ is the ratio of working to intake airflow rate. Each wall plate is $\delta_{wall}=0.375$ mm thick and is composed of plastic film and cellulose fibre sheet. The water film thickness is assumed to be $\delta_{water}=0.3 \text{ mm}$. The plates are L=
0.86 long in the flow direct and W= 0.458 wide. The plate separation distances, i.e., the channel height and width are 3.0 mm and 45.8 mm respectively. There are N=320 pairs of channels.

Fig. 1 The plate type heat and mass exchanger in a counter-flow configuration

2.2 Governing equations and numerical solution method

An efficient numerical model of the heat exchanger can be obtained by applying the modified ε-NTU solutions to each sub-heat exchanger and requiring that the boundary conditions associated with each sub-heat exchanger are consistent with the adjacent sub-heat exchangers. The following equations are coded in Engineering Equation Solver (EES) software and solved by Newton's alternative method. It is found that the heat conductance approach constant rapidly when the number of computational elements is larger than 5, therefore, 30 computational elements number was used to provide sufficient accuracy. A few assumptions are made to simplify the solutions: 1) the heat exchanger is well insulated; 2) axial heat conduction and fouling resistance of heat exchanger is ignored; 3) In each computational sub-heat exchanger, heat and mass transfer coefficients as well as fluid properties are fixed; 4) the Lewis number is unity. 5) the temperature of air and water interface is saturated at the temperature of water film.

The governing mathematical equations for each sub-heat exchanger, or computational element, are given as follows. The heat transfer rate for product air in dry channels is expressed by Eq. (1)

\[
dq = \frac{V}{p} (i_{p1} - i_{p2})
\]

The heat transfer rate for working air in wet channels is given by Eq. (2)

\[
dq = \frac{V}{w} (i_{w1} - i_{w2})
\]

For the regenerative configuration, the outlet air conditions in dry channel is equivalent to the inlet air conditions in wet channels, i.e., \(t_{p,o}=t_{w,i}\) and \(t_{p,o}=t_{w,i}\).

The heat transfer between product air and working air is given by Eq. (3)

\[
dq = UdA (i_s(t_p) - i(t_w))
\]

Where, the overall conductance of the heat exchanger is determined by Eq. (4)

\[
UdA = \left[ a \left( \frac{1}{h_p} + \frac{1}{h_w} + \frac{1}{k_{wall}} + \frac{1}{k_{water}} \right) + \frac{1}{h_m} \right]^{-1} NWdx
\]

It is assumed that the air enthalpy has a linear relation with saturation temperature, which is defined as Eq. (5).

\[
i_s(t_p) = a(t_p) + c
\]

The heat capacity rate of product air and working air can be redefined by establishing the heat balance on product air and working air, which are expressed as the following two equations.

\[
\frac{V}{p} c_{p,p} (t_{p1} - t_{p2}) = C_p \left( i_s(t_{p1}) - i_s(t_{p2}) \right)
\]

\[
\frac{V}{w} c_{w,w} (t_{w1} - t_{w2}) = C_w \left( i(t_{w1}) - i(t_{w2}) \right)
\]

Rearranging Eq. (6) gives:

\[
C_p = \frac{\frac{V}{p} c_{p,p} (t_{p1} - t_{p2})}{i_s(t_{p1}) - i_s(t_{p2})} = \frac{\frac{V}{p} c_{p,p}}{a}
\]
Rearranging Eq. (7) gives:
\[ C_w = \frac{V_w c_{w,p} \left( t_{w1} - t_{w2} \right)}{i(t_{w1}) - i(t_{w2})} = \frac{V_w}{i(t_{w1}) - i(t_{w2})} \tag{9} \]

As the overall conductance approaches infinity, the heat exchanger could obtain the maximum possible heat transfer rate:
\[ dq_{\text{max}} = C_{\text{min}} \left( i(t_p) - i(t_w) \right) \tag{10} \]

Where, \( C_{\text{min}} = \min(C_p, C_w) \)

The effectiveness of sub-exchanger is the ratio of the actual heat transfer rate to the maximum possible heat transfer rate:
\[ e = \frac{dq}{dq_{\text{max}}} \tag{11} \]

The number of transfer units is given by Eq. (12):
\[ NTU = \frac{UdA}{C_{\text{min}}} \tag{12} \]

For the heat exchanger in counter-flow configuration, the effectiveness is expressed by Eq. (13):
\[ e = \frac{1}{1 - C_r e^{-NTU/C_r}} \tag{13} \]

Where, \( C_r = \frac{C_{\text{min}}}{C_{\text{max}}} \) and \( C_{\text{max}} = \max(C_p, C_w) \)

For each computational element, the local convective heat transfer coefficients from product air and working air to exchanger walls are determined by Eqs. (14) and (15) respectively:
\[ h_p = \frac{N_{p} k_p}{D_p} \tag{14} \]
\[ h_w = \frac{N_{w} k_w}{D_w} \tag{15} \]

For a uniform wall temperature, the local Nusselt number for a laminar, hydrodynamically and thermally fully developed flow in a rectangular duct \((b/e=0.066)\) is given by Eq. (16).
\[ Nu = 7.54 \tag{16} \]

The mass transfer coefficient between working air and water film can be determined by the following formula:
\[ h_m = h_w \frac{c_{p,w}}{w} \tag{17} \]

For laminar flow in rectangular duct, the average friction factor of each element is given by:
\[ f = \frac{96}{\text{Re}} \tag{18} \]

Wet-bulb effectiveness of the heat exchanger is defined as the following equation:
\[ e_{wb} = \frac{t_{pi} - t_{wo}}{t_{pi} - t_{wb,pi}} \tag{19} \]

The sensible cooling capacity is determined by Eq. (20):
\[ Q_{\text{cooling}} = \frac{c_p(V_p - V_w)(t_{pi} - t_{po})}{3.6} \]  

(20)

Coefficient of performance (COP) is defined by the ratio of sensible cooling capacity to total power consumption by fan and water pump of the cooler as below:

\[ COP = \frac{Q_{\text{cooling}}}{P_{\text{fan}} + P_{\text{pump}}} \]  

(21)

3. Results and analyses of numerical modelling

Fig. 2 plots the effects of inlet air temperature on wet-bulb effectiveness, product air outlet temperature, cooling capacity and COP of the exchanger under three different humidity ratio values. The prediction results show that, for the fixed air humidity ratio, the product air outlet temperature, cooling capacity and COP of the cooler increased with increasing the product air inlet temperature from 25 to 45 °C. However, the wet-bulb effectiveness didn’t alter significantly as the inlet air temperature varies. As the humidity ratio of inlet air reduces from 16 to 6 g/kg, the cooler’s cooling capacity and COP were increased significantly. Meanwhile, the wet-bulb effectiveness didn’t change too much. The findings reveal that the cooler would be more suitable for applications in hot and arid regions than hot and humid regions. For applications in hot and humid regions, air dehumidification unit should be added prior to the evaporative cooler to reduce air humidity.

![Fig. 2](image)

Fig. 2 The effect of inlet air temperature on (a) wet-bulb effectiveness, (b) product air outlet temperature, (c) cooling capacity and (d) coefficient of performance of the counter-flow heat and mass exchanger

4. Conclusions

This paper develops a numerical mode of regenerative evaporative cooler in counter-flow configuration based on modified \(\varepsilon\)-NTU method. The effects of inlet airflow temperature and humidity were predicted by the model. A few conclusions are summarised as follows: The inlet air temperature has positive impacts on cooling capacity and COP. As the inlet humidity increases, both the cooling capacity and
COP reduce while the outlet temperature of product air increases; The wet-bulb effectiveness changes minorly as either inlet air temperature or inlet air humidity vary significantly.

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Nomenclature:

- \( a \) — the slope of the temperature-enthalpy saturation line
- \( c \) — constant in equation
- \( c_p \) — specific heat capacity at constant pressure, \( J/(kg\cdot K) \)
- \( C \) — heat capacity rate, \( (J/K) \)
- \( dq \) — heat transfer rate between product and working air for each computational element, \( W \)
- \( D \) — hydrometric diameter of dry or wet channel, \( m \)
- \( h \) — convective heat coefficent, \( W/(m^2\cdot K) \)
- \( h_m \) — mass transfer coefficicent, \( kg/(m^2\cdot s) \)
- \( i \) — enthalpy of airflow, \( J/kg \)
- \( k \) — thermal conductivity, \( W/(m\cdot K) \)
- \( NTU \) — number of transfer units
- \( Nu \) — Nusselt number
- \( P \) — power consumption, \( W \)
- \( Q \) — cooling capacity of the cooler, \( W \)
- \( UdA \) — overall conductance of the heat exchanger, \( W/K \)
- \( V \) — volumetric airflow rate, \( m^3/s \)

Greek symbols

- \( \varepsilon \) — wet-bulb effectiveness
- \( \rho \) — density of airflow, \( kg/m^3 \)
- \( \sigma \) — surface wettability factor, \( 0.8 \)

Subscripts

- \( i \) — inlet
- \( o \) — outlet
- \( p \) — product air
- \( w \) — working air
- \( wb \) — wet-bulb
- \( s \) — saturation
- \( 1 \) — inlet for each computation element
- \( 2 \) — outlet for each computation element
- \( \max \) — maximum
- \( \min \) — minimum
- \( \text{fan} \) — air fan
- \( \text{pump} \) — water pump

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