Simulation for evaporative cooling in partially wetted plate heat mass exchanger

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Abstract. The thermal characteristics of a M-cycle counter flow type plate heat exchanger (PHE), intended for use as a cooler, has been investigated numerically as an indirect evaporative cooler. For a fixed heat exchanger length \( L = 50d \) and the space between plates \( d = 6 \text{ mm} \), the investigation included a numerical simulation for the PHE and the study of its performance with the variation in the number of the wetted zone \( n \) from 1 to 16 for inlet Reynolds number \( \text{Re} = 200 \). The computational results indicate that the heat exchanger performs better with the number of the wetted zone, high inlet air temperature and low relative humidity. Results show that the cooler consumes less water and it is an environmentally friendly cooler for dry areas.

1. Introduction
Indirect evaporative cooling (IEC) involves cooling an airstream and keeping its water content unchanged. IEC systems are increasingly used for comfort cooling in the dry and temperate climates. The resulting indoor condition with IEC would be at a lower relative humidity than that obtained with direct evaporative cooling (DEC). This is because the IEC reduces the process air (primary) temperature without increasing the moisture content (constant moisture content or constant dew point temperature) whereas, in the case of DEC, air is being moistened at a constant wet-bulb temperature to reduce its dry bulb temperature. The enthalpy of the primary airstream decreases because no moisture is added to it. This process contrasts with direct evaporative cooling, which is essentially adiabatic [1–4]. The usefulness of indirect evaporative cooling is related to the depression of the wet-bulb temperature of the secondary air below the dry-bulb temperature of the entering primary air. Therefore the secondary air stream is evaporatively cooled rather than the primary air stream. Heat and mass transfer processes through air-water interfaces are of a major importance in many engineering applications.

The present study deals with numerical analysis of a flow and heat and mass transfer for indirect evaporative cooling of a M-cycle counter-flow heat exchanger that represents a working dry channel, product channel and wet channel with plane-parallel partially wetted walls. Particular attention is given to influence of the parameters studied (the number of the wetted zone, inlet air temperature, inlet relative humidity \( \phi_b \)) on the latent heat transfer, wet bulb effectiveness and humidification effectiveness.
2. Analysis and modeling
The problem is schematically shown in figure 1. This cooler consists of several plates that are designed to wick water evenly on one side and transfer heat through the other side. The plates are stacked on each other. The working and product air flow parallel to the channels on the dry side, and then the working air enters the wet side to allow the working air to be pre-cooled before entering the wet side by losing heat to the opposite wet surface. This pre-cooled air flows over the wet surface along with channels absorbing heat from the working and product air. Thus, the working air is humidified, heated and discharged to the atmosphere, while the product air is cooled before being delivered to spaces. The cooler is heat-insulated from outside ($q_w = 0$). The thickness of the liquid film on the walls of wet channel is assumed infinitely small and it does not affect the fluid flow in the channel and the thermal resistance. All parameters at the inlet ($u_0, T_0, \phi_0$) were constant along channel height. This study considers a laminar stationary flow without regard to radiant heat transfer and viscous dissipation, neglecting the Dufour and Soret effects.

The main equations for the laminar forced convection are written as: continuity equation; motions in the $x$ and $y$ direction; energy equation; water vapor diffusion equation.

The system of differential equations along with the boundary conditions was solved numerically using the finite volume method. To relate velocity and pressure, we used the SIMPLE algorithm. For discretization we used uniform meshes whose optimal size was chosen in a special series of calculations and was $200 \times 120$ in longitudinal and transverse directions, respectively. A 2-D fluid flow and heat transfer numerical analysis for the counter flow PHE as an indirect evaporative cooler was carried out by writing a FORTRAN program [5]. Test correlations on the fluid flow and heat transfer in channels with adiabatic and isothermal walls without phase transitions have demonstrated good agreement between results of the present calculation and the data of [6].

3. Results and discussion
For a fixed heat exchanger length ($x/d = 50$) and the space between plates ($d = 6\text{mm}$), figure 2 shows effect of the number $n$ of the humid zones on the air temperature distribution for inlet $Re = 200$, $T_0=30^\circ\text{C}$ and $\phi_0 = 0$. For a working wet channel with fully wetted walls ($n=0$), in this flow regime the main changes of temperature occur mainly in the inlet section of wet channel. In both working dry channel and product channel, the air is cooled by the air in adjacent working wet channel so that the air temperature decreases along the channel length. In the working wet channel, the working air temperature increases along the air flow direction. At a certain length of all the channels, the lowest air temperature is obtained in the working wet channel so that the heat is transferred from dry channels to adjacent working wet channel. The heat is absorbed (latent heat) due to water evaporation in the working wet channel. The air in both working dry channel and product channel is cooled without humidity change. The temperature distribution is highly influenced by the presence of the dry zone.
As can be seen from figure 2 $(n = 1)$ and figure 3, the outlet air temperature of product channel is maximum and concentration at the exit of wet channel is minimum for $n = 1$. For a working wet channel with partially wetted walls as shown in figure 2 $(n = 4$ and $16)$, the air temperature of the wetted zones near the heat clapboard is lower than temperature in the center of the channel. It is because the wet zone makes the temperature less on surface layer of the heat clapboard. The thermal gradient of the process is much more complicated and takes considerably longer distance from the entrance of wet channel. A similar change takes place for the concentration fields, this is due to the fact that the presence of the dry zone produces a higher wall temperature with an accompanying increase of the wall partial pressure of saturated vapor for the next wet zone. At the same time, the presence of dry zones induces a decreasing relative humidity of the working air in the wet channel and the cooler consumes less water than fully wetted walls.

The wet bulb effectiveness is defined as the ratio of the temperature difference between the inlet and outlet product air to the difference between the inlet product air dry bulb and inlet working air wet bulb temperature. This reflects the extent of the approach of the outlet product air temperature of the IEC against the wet-bulb temperature of the inlet working air, and can be written as:

$$\varepsilon = \frac{T_{0,\text{dry}} - T_{\text{w3, out}}}{T_{0,\text{dry}} - T_{\text{wet bulb}}}.$$

The humidification effectiveness is defined as the ratio of the moisture content difference between the inlet and outlet working air to the difference between the moisture content of saturated air and inlet working air moisture content. This reflects the extent of the approach of the outlet working air moisture content against the moisture content of saturated air of the inlet working air related to the IEC, shown as follows:

$$\psi = \frac{K_w - K_{0,\text{wet}}}{K_T - K_{0,\text{wet}}}.$$

where $K_T$ is concentration of water vapors for saturated air. The calculation results of thermal and humid efficiency depending on the number of wet zones are represented in figure 3 (Re = 200, $T_0 = 30^\circ$C, $\varphi_0 = 0$).
Figure 3. The effect of $n$ number on effectiveness and product air parameters.

For a fully wetted channel ($n = 0$) the bulk concentration at the exit of wet channel is maximum and $\varepsilon = 0.88$, $\psi = 0.44$. Minimum efficiency occurs for the case with one wet and one dry zone ($n = 1$). However, this decrease is not so essential and with an increase in the number of wet zone ($n$) the value of effectiveness rises, approaching the asymptotic value $\varepsilon \to 0.8$, $\psi \to 0.4$.

4. Summary
A numerical analysis has been carried out to investigate the heat and mass transfer characteristics of counter-flow multi-channel indirect evaporative cooler. The parameters that affect the thermal performance of the cooler have been examined. The results show that the increase in the number of wetted zone leads to an increase in the parameters of thermal and humid efficiency. The minimum value of these parameters is observed when $n = 1$, however, for the considered conditions ($Re = 200$) the suppression of the evaporation processes is not large. These findings are of significance for the design of an indirect evaporative air cooler and the results show that the cooler consumes less water and it is an environmentally friendly cooler for dry areas. The obtained data can be used for optimization analysis of air cooling with variation of Reynolds number, heat exchanger length and plate spacing.

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