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# Numeric study of heat and moisture transfer process in channels of a dew point evaporative cooler

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## ABSTRACT

The heat and moisture transfer efficiency is the key factor that affects the energy efficiency of an evaporative cooling system. The cooling of air in a wet channel is a complex process. It is necessary to find the dominant influences of this cooling process and increase the energy saving potential. This paper develops a validated computing model under the EES (Engineering Equation Solver) environment, and numerically explores the heat and moisture transfer process in channels of a dew point evaporative cooler. The results show that it always achieves the highest evaporation efficiency within 0.4m from the entrance of the wet channel. That is to say, the effective length of the exchanging channel could be significantly shortened in contrast to the popular products widely reported.

**Keywords:** evaporative cooling, dew point air cooler, heat and moisture transfer, evaporation rate

## NONMENCLATURE

| | |
|---|---|
| $t$ | temperature, ºC |
| $C_p$ | specific heat capacity, kJ/kg·ºC |
| $h_m$ | mass transfer coefficient, m/s |
| $h$ | convection coefficient, W/(m$^2$·ºC) |
| $Re$ | Reynolds number, - |
| $Nu$ | Nusselt number, - |
| $l$ | length, m |
| $Q$ | mass flow rate, kg/s |
| $d$ | moisture content, g/kg dry air |
| ***Subscripts*** | |
| w | water film |
| p | dry channel |
| s | wet channel |

## 1. INTRODUCTION

In recent years, more and more scholars have conducted research on dew point evaporative cooling air conditioners in two main directions. One is to develop prototypes and carry out experimental measurements on it. Xu [1, 2] has developed a super performance dew point air cooler, which achieved a COP as high as 52.5 and 3 times more cooling capacity than a commercial dew point air cooler of the same size. The other is to optimize the structure of a product by simulation work. In order to improve the cooling efficiency and energy effectiveness, many scholars [3-5] studied the cooling effect of air in the channels under different parameters, i.e. channel length, temperature, humidity, air speed and working air ratio. The results show that under local climatic conditions, the wet bulb efficiency can reach as high as 120~130 and the dew point efficiency for 80~90%.

The cooling of air in a wet channel is a complex process. In order to find the dominant influences of this cooling process and increase the energy saving potential, simulations were conducted to obtain changes in the operation parameters (e.g., temperature, humidity, evaporation rate, etc.).

## 2. MODEL SET UP

### *2.1 Physical model*

The heat exchanger of the dew point evaporative cooler is arranged alternately by several dry and wet channels with equal geometric conditions, and the computational

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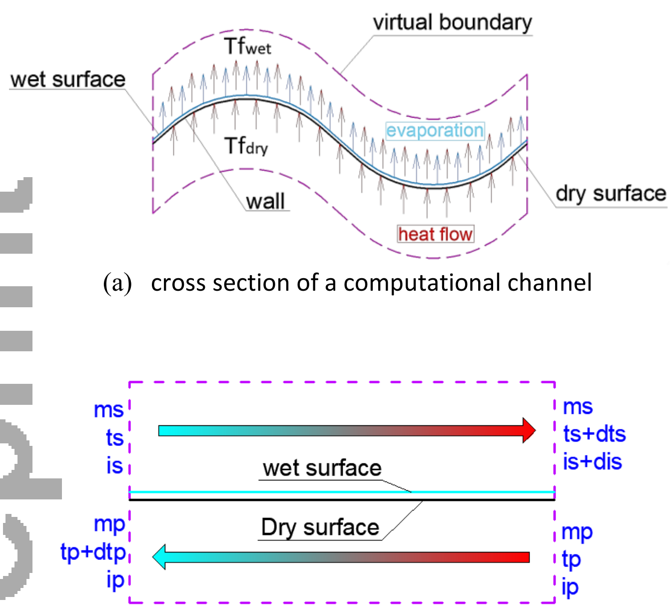
channels are coupled with each other. The "finite element" method is used to "differential" the computational channels and establish the relevant differential equations. Consider the computational element as the basic unit of the simulation, and the finite elements constitute a computational channel, see Fig. 1. The Newtonian iterative method is applied to investigate the equilibrium state in each channel from the perspective of heat and mass transfer.




(a) cross section of a computational channel

(b) computational model

Fig. 1 Computational channel.

To simplify the simulation process and mathematical analysis, the following assumptions have been made:

(1) The boundary surface of the heat exchanger is adiabatic.

(2) The heat and mass transfer process inside the channel is steady state.

(3) The heat transfer through the channel wall is in the vertical direction. Only convective heat transfer is considered between the airflow and water film, and between the airflow and the channel wall, ignoring the influence of natural convection.

(4) The surface temperature of the wet channel is equal to the water temperature at the same moment.

(5) The thermal resistance of the channel wall is negligible, i.e. there is no temperature difference between the wet and dry channel surfaces.

(6) The air in a computational channel is incompressible, and the air velocity is uniform.

*2.1 Mathematical model*

Heat transfer in dry channel well:

$$\frac{\partial t_p}{\partial y} = \frac{K}{C_{p,p}} \frac{A}{m_p l_p} (t_w - t_p) \tag{1}$$

Mass transfer in wet channel well:

$$\frac{\partial d_s}{\partial x} = \frac{h_m A}{m_s l_s} (d_w - d_s) \tag{2}$$

Heat transfer in wet channel well:

$$\frac{dQ_s}{dx} = \frac{A}{m_s l_s} [h_s (t_w - t_s) + r_w h_m (d_w - d_s)] \tag{3}$$

Energy balance in dry/wet well:

$$\frac{dQ_w}{dx} = -\frac{C_{p,w}}{l_p} \left(m_w \frac{\partial t_w}{\partial x} + t_w \frac{\partial m_w}{\partial x} + \frac{\partial m_w}{\partial x} \frac{\partial t_w}{\partial x} dy\right) dy \tag{4}$$

Energy balance for the coupled dry & wet elements:

$$\frac{\partial t_w}{\partial x} = -\frac{l_p}{C_{p,w} m_w} [K_p (t_w - t_p) + h_s (t_w - t_s) + r_0 h_m (d_w - d_s)] \tag{5}$$

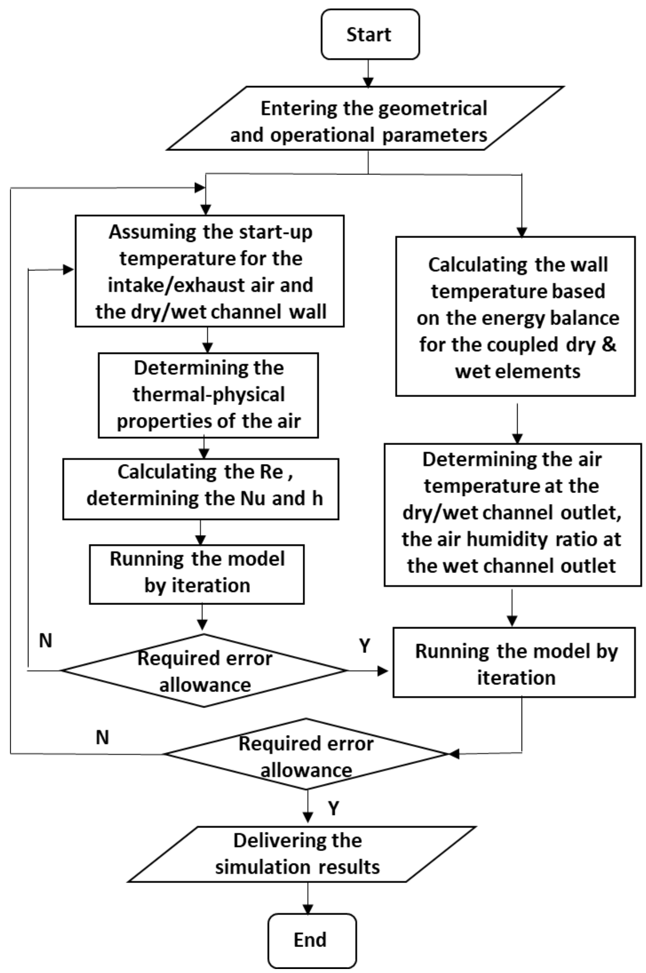



Fig. 2 Calculation process chart

*2.2 Calculation process*

To enable simulation of the heat and mass transfer processes occurring in the different elements/channels, a dedicated computational algorithm is developed to solve the above equations using the finite element and Newton-iteration method, which is operated under the EES (Engineering Equation Solver) environment. The Newton-iteration algorithm [6] used for developing the computer model is shown in Fig. 2.

## 3. MODEL VALIDATION

Calibration and validation of the computational model is necessary to achieve a reasonable fit between the model output and the actual prototype. Simulations were performed based on published experimental results[7]. Given a channel spacing, length and width of 5 mm, 1200 mm and 80 mm, respectively, an inlet air velocity of 2.4 m/s and a mass to inlet air ratio of 0.33, i.e., the experimental conditions applied in [7], the simulations were performed to produce a series of modeling results. Therefore, the following validation is based on the product air temperature. The simulation results were compared with the testing data to examine effectiveness and accuracy of the model.

By analyzing the water temperature variation between the outlet and inlet, as shown in Fig. 3, the simulation results are consistent with the reference experimental data with high accuracy. The ideal trend consistency and reasonable deviations indicate that the simulation results are reasonably reliable.

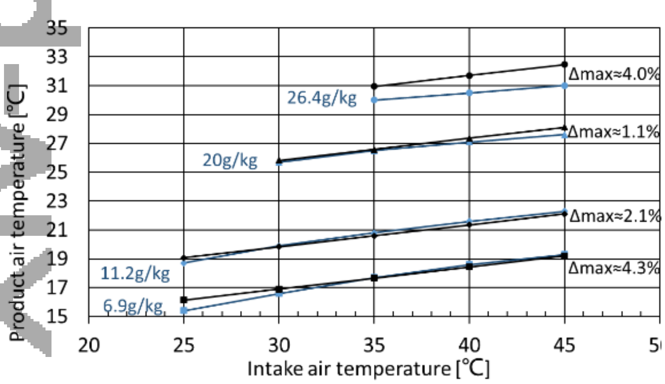



Fig. 3 Calculation process chart

## 4. SIMULATION RESULTS AND ANALYSIS

To further validate the proposed simulation model and investigate the heat and mass exchanging process inside the channels, a series of simulation results was taken to be analyzed. The following analysis will focus on the internal heat and mass transfer process along the channels.

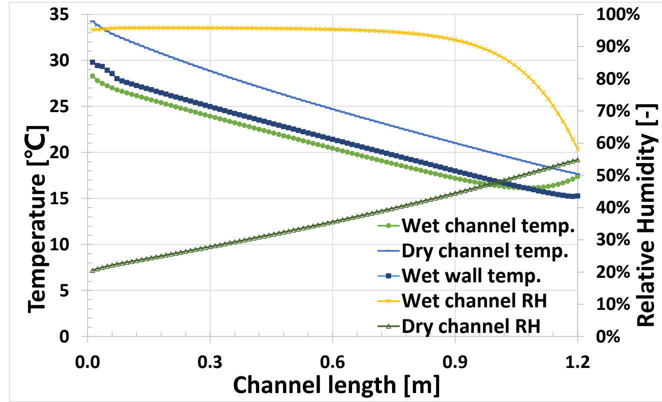



Fig. 4(a) Variation of temperature and RH along the channels.

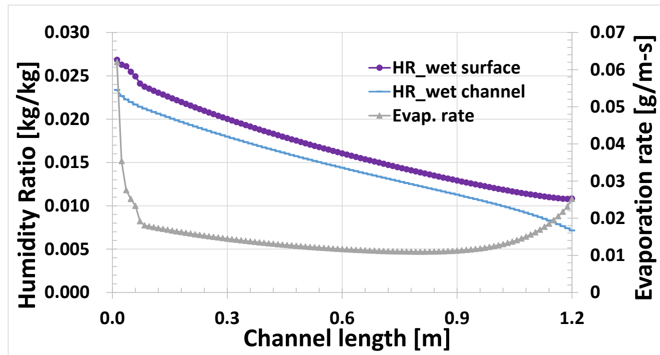



Fig. 4(b) Variation of HR and evaporation rate in the wet channel.

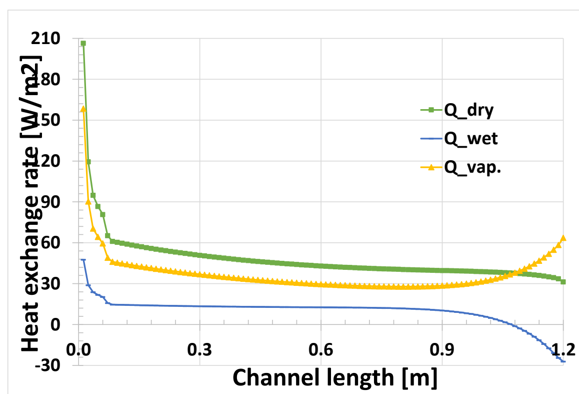



Fig. 4(c) Heat exchange along the channel length.

Fig 4: Heat and mass exchange progress in the channels.

Fig. 4(a)-(c) presents the dynamic processes of the parameters, i.e. the temperature, relative humidity (RH), humidity ratio (HR) of the air, the temperature of wet surface, the evaporation and heat exchange rate along the dry and wet channel length, based on the dedicated numerical simulation. It should be noted that the processes in dry and wet channels are in opposite direction for the counter-flow pattern, i.e. the start point of the dry channel corresponding to the end point of the wet channel and vice versa. Therefore, the process in dry channel starts at the channel length of 0 while the process in wet channel starts at the channel length of 1.2m.

Fig. 4(a) shows nothing special along the dry channel, i.e. the relative humidity increases with the air temperature drop. In the wet channel, the air temperature keeps decreasing at the short initial stage within around 0.1m due to the effect of the low temperature entering water, and then maintains increasing along the wet channel being heated by the air in adjacent dry channels. In the extent about 0.4 m from the start point, the air relative humidity in the wet channel rises rapidly from 60% to 95% owing to the water evaporation, then in the rest stage, the relative humidity keeps close to saturation with the evaporation process and the increase of air temperature. The temperature of the wet wall, equaling to the temperature of the water as pre-assumed, keeps ascending with the heat exchanging process with the air in the adjacent dry channel.

Fig. 4(b) shows the variation of humidity ratio and evaporation rate in the wet channel along the channel length. The driving force exists in the difference of partial vapour pressure/humidity ratio between the wet surface and the inner air. With the temperature increase, the humidity ratio at saturated state over the wet surface keeps rising and humidity ratio of the wet air ascends due to the process of evaporation. As for the slight air temperature drop in the initial 0.1m (Fig. 4(a)), which affects the partial vapour pressure at the same humidity ratio, it could be seen that the evaporation rate reduces while the HR difference even more. Comparing the variation trends between the evaporation rate and HR difference, it could be concluded that the temperature predominates the evaporation process superior to the HR difference in such case.

Fig. 4(c) demonstrates the heat exchanging process along the channel length. $Q$_dry means the heat exchange between the airflow and the wall in the dry channel, $Q$_wet for heat exchange between the airflow and the wall in the wet channel and $Q$_vap for the heat transfer by water evaporation. Within the initial stage of about 0.1 m in the wet channel, the heat flows from the air to the wet channel wall due to the low temperature of entering water and thus $Q$_dry presents a minus value. Owing to the relatively low evaporation rate in the extent of 0.2 m, $Q$_vap shrinks till the evaporation reaches a normal state with the wall temperature increase. $Q$_dry is the result of the sum of $Q$_wet and $Q$_vap.

## 5. CONCLUSIONS

(1) In the range of 0.4m from the entrance, due to the evaporation of water, the relative humidity of the air in the wet channel rises from 60% to 95% rapidly, then the change slows down, and finally keeps constantly close to the saturation.
(2) In the first 0.1m of the wet channel, the air is rapidly cooled by the wall of the wet channel due to the low relative humidity and the low temperature of the circulating water.
(3) The prototype could be economically manufactured by shortening the heat transfer plates and the air can be more effectively cooled in the channels. The cooling load can be met by various splicing modules in steps.

## ACKNOWLEDGEMENT

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## REFERENCE

[1] Xu P, Ma X, Zhao X, Fancey KS. Experimental investigation of a super performance dew point air cooler. Applied Energy,2017;203:761–777.
[2] Xu P, Ma X, Thierno M.O., Zhao X, Fancey K. Numerical investigation of the energy performance of a guideless irregular heat and mass exchanger with corrugated heat transfer surface for dew point cooling[J]. Energy 2016; 109:803–817.
[3] ZHAO X, LI J M, RIFFAT S B. Numerical study of a novel counter—flow heat and mass exchanger for dew point evaporative cooling [J]. Applied Thermal Engineering， 2008，28(14) : 1942—1951.
[4] X.Cui, K.J. Chua, W.M. Yang,K.C. Ng,K. Thu,V.T. Nguyen. Studying the performance of an improved dew-point evaporative design for cooling application[J]. Applied Thermal Engineering, 2014,63(2).
[5] M. Jradi,S. Riffat. Experimental and numerical investigation of a dew-point cooling system for thermal comfort in buildings[J]. Applied Energy,2014,132.
[6] Riangvilaikul B, Kumar S. An experimental study of a novel dew point evaporative cooling system. Energy Build 2010; 42:637-44.
[7] R. Boukhanouf, H. G. Ibrahim, A. Alharbi, and M. Kanzari. Investigation of an evaporative cooler for buildings in hot and dry climates. Journal of Clean Energy Technologies, Vol. 2, No. 3, July 2014.