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Mathematical Modelling of Indirect Evaporative Cooler Having Different Types of Flow Configurations

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ABSTRACT

Air conditioning and refrigerating systems consumes about 15% of total electricity production of the world. The heating, ventilating and air conditioning load of the world is estimated to rise 6.2% per year. Conventional mechanical compressor systems which occupies about 95% of cooling systems are characterized by high consumption of electricity. Evaporative cooling is one method of utilizing natural cooling effect to cool the building. In an indirect evaporative cooling method the air is cooled significantly without an increase in its humidity ratio. This paper presents mathematical modelling of heat and mass transfer in indirect evaporative coolers with three flow configurations: parallel flow, counter flow and regenerative. The overall balance equations and boundary conditions for three flow configurations of indirect evaporative cooling system are derived.

Keywords—Indirect evaporative cooling, Mathematical model, Heat and mass transfer

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I. INTRODUCTION

Buildings contribute to about 20-40 percent of total energy consumption of the world. In buildings sector Heating ventilation and air conditioning (HVAC) systems consume more than half of energy consumed by buildings i.e. about 10-20 percentage of total energy [1]. Increasing global temperatures, demand for more comfort and increase of buildings area results in great increase in HVAC energy consumption of the world. Therefore, it is necessary to improve efficiency and decrease environmental impact of HVAC systems. Conventionally, Mechanical compressor based refrigeration systems are used in air conditioning applications. Evaporative air coolers own such approval to the fact that they are able to reduce significantly the energy used for air conditioning systems. These devices have very high COP of about 200, which is substantially higher compared to typical vapour-compression systems. Considering above facts, evaporative cooling systems are the most popular alternatives of conventional air conditioning systems. Actually, evaporative cooling had its birth around one thousand years ago invented in ancient Egypt. At that time, porous pots and ponds covered with a wet cloth were often used to preserve food against hot weather and some water chutes were also integrated into walls to keep the inside space cool [3], due to the evaporation of water when air flowed through. An evaporative cooling system is more feasible in hot and arid climatic conditions in terms of thermal performance owing to the notable potential of humidity increment of air which results in significant temperature reduction. Evaporative cooling is the process of cooling the hot and dry air by removal of heat by vaporization of water that comes in contact with it increasing the relative humidity of dry air. The

evaporative cooling system can be classified as direct or indirect contact with regard to interaction between the streams. Direct evaporative coolers (DEC) were first used in air conditioning applications. Direct evaporative cooling system has been theoretically and experimentally studied by many scholars due to its easy fabrication and high efficiency in hot and dry districts [9-12]. Those devices were largely developed in the dry climate regions of the world [2]. In DEC, the cool air thus obtained has high relative humidity which is not suitable for air conditioning applications. However, if the outdoor dry bulb temperature (DBT) is not high enough or outdoor humidity ratio (HR) is not low enough, the DEC cannot cool the air nor water down to the required level of temperature.

Indirect evaporative cooler (IEC) is an energy efficient and environmental friendly cooling device which uses water evaporation to produce cooling air. It receives increasingly attention in the field of building energy conservation for its high efficiency, good comfort, pollution-free and easy maintenance features [4-8]. IEC is used to cool the air without increase in its humidity ratio. IEC separates primary air from sprayed water by installing dry and wet channels. Thus primary air is delivered in the dry side of the heat exchanger, meanwhile secondary air also known as the working air flows across the wet side, in a direct contact with sprayed water. Through the process, heat released from primary air is transferred to the wet channel and then absorbed by the water film covered at wet surface. The resulted water evaporation is taken away by the secondary air which then is discharged to the ambient The indirect evaporative cooler (IEC), regarded as a low-carbon cooling device, was proposed as fresh air pre-cooling and heat recovery device in the air-conditioning system to break the region limitation of application in hot and humid areas. In this hybrid system, the exhausted air with low temperature and humidity from air-conditioned space is used as secondary air to cool the inlet fresh air. As the dew point temperature of the fresh air is high, condensation may occur in the dry channels.

The evaporative cooler was an interesting subject in US patents in the beginning of 20th century. There wasn't any significant attempt for analysing the DEC before the study reported by Watt [3]. He presented the advantages and disadvantages of the direct evaporative cooling systems. Dai and Sumathy [10] studied a numerical model of DEC with equations for the liquid film, the liquid-gas boundary and the gaseous phase. They indicated that the performance of the DEC system can be changed by variation in some physical parameters of the air cooler. Camargo used both investigational and mathematical results of the DEC test with rigid cellulose media. Their results showed that efficiency of DEC increases with decrease in air velocity [9]. Hosseni and Beshkani examined the parameters that effect the saturation efficiency of DEC [13]. They used corrugated paper to increase the cooling media up to the moistened surface area of 400Sq.m. Rajput and Kulkarni studied the theoretical performance of the DEC with diverse pad materials. They presented that material with greater wetted surface area and lower air mass flow rate gives higher saturation efficiency [14]. Sheng and AgwuNnanna investigated the cooling performance of a DEC by evaluation of the speed and the dry-bulb temperature of inlet air and moreover the effects of the inlet water temperature [11].Jaber [17] installed and analyzed an indirect evaporative cooling system in Jordan which perfectly represents the climate of Mediterranean. With the operation of this system, the energy consumption and emission of carbon based gases were significantly reduced without influencing the comfort conditions. According to the data, if 500,000 Mediterranean buildings use indirect evaporative cooling system instead of conventional air conditioning, every year about 1084GWh/a energy can be saved and 637,873 ton emission of CO2 would be reduced. Still it took less than two years to get the payback.Hasan presented the analysis of the novel indirect evaporative air cooler based on the modification of the e-NTU method, the wet-bulb effectiveness achieved in this study was up to 1.2 [15].

The above literature review indicates the attractiveness of indirect evaporative cooling systems. The main objective of this study is to synthesize a numerical model of an IEC.

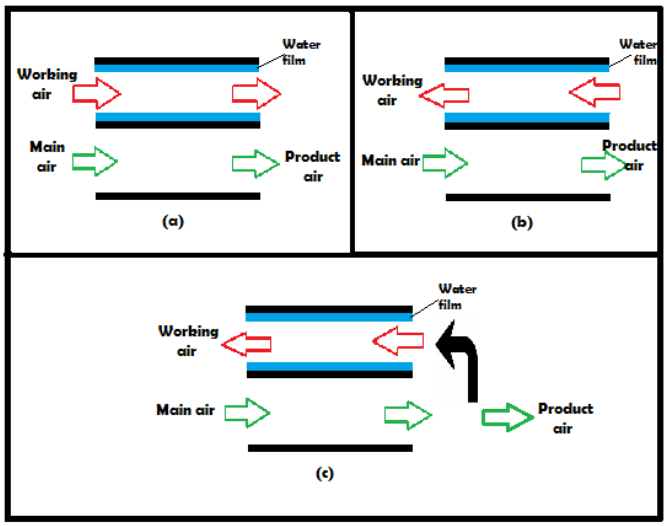

Fig.1. Main flow configurations of indirect evaporative air cooling: (a) parallel-flow (b) counter-flow (c) regenerative

II. MATHEMATICAL MODELLING

Non-adiabatic evaporation in the IEC can be described by four processes. Heat and mass transfer between water film and secondary air stream, heat transfer between water film and walls, longitudinal heat conduction in the walls and heat transfer between walls and primary airstream. The main objective of this study is to investigate the IEC with three flow configurations (parallel, counter and regenerative) to realize the advantages and limitations of each flow configuration to find paths to their improved performance. The study is conducted with the mathematical model of heat and mass transfer processes. These models can be created using Reynold's equations, boundary layer equations and ε-NTU method. Due to the complicated air flow and water distribution of the models based on Reynold's equations and boundary layer equations, they are less appropriate to solve modern IEC. The presented mathematical model is based on a modified ε-NTU method in which air flow is considered as fluid with constant temperature, mass transfer potential and velocity which are equal to bulk average values in normal sections to plates of exchanger. The assumptions for the presented mathematical model:

1. The process is steady-state.
2. The system is isolated and no internal heat generation.
3. Constant specific heat of the fluid, vapor and air.
4. Air assumed to be an incompressible gas.
5. Spray water is circulated in a closed-loop.
6. There is no conduction in the water layer in the flow directions.
7. Air flow heat capacity is much larger than that of the water
8. Temperature of the air and water in normal section of the flow is uniform.

The heat and mass schematics of control volumes in the dry and wet channels, which demonstrate operating principle of mathematical model is presented for parallel flow IEC in Fig 2.The heat and mass transfer processes in other IECs are analogous to that of parallel flow IEC.

The full derivation of heat transfer balance equations for indirect evaporative air coolers can be seen in [16]. The numerical model is based on the mass and heat transfer equations given below.

1. Energy balance equations for main air flow and working air stream.
2. Mass conservation equations for water.
3. Energy balance for the fins in both wet and dry channels.
4. Equations for airflow-surface interface in both dry and wet air flow passages.

For the main air flow, the energy balance equation in parallel, counter and regenerative IEC is identical. The energy conservation equation for main air flow in parallel, counter and regenerative IEC:

$$\frac{dT}{d\bar{x}} = NTU_1{}^a \begin{bmatrix} \frac{2}{m^{fin1}s^{fin1}} tanh(h^{fin1}m^{fin1}) + \\ \left(1 - \frac{\partial^{fin1}}{s^{fin1}}\right) \end{bmatrix} (T'^{\text{plate1}} - T_1) \quad (1)$$

The energy conservation equation for working air stream:

- For parallel flow IEC:

$$\frac{dT}{d\bar{x}} = NTU_2{}^a \begin{bmatrix} \frac{2h^{fin2}}{s^{fin2}} \int_{\bar{z}_2=0}^{\bar{z}_2=1} [T'^{\text{fin2}} - T_2]\, d\bar{Z}_2 + \\ \left(1 - \frac{\partial^{fin2}}{s^{fin2}}\right) (T'^{\text{plate2}} - T_2) \end{bmatrix} (2)$$

The energy conservation equations for counter and regenerative IEC are similar to that of a parallel flow IEC, but they are characterized by different sign, due to opposite direction of air flow.

- For counter, regenerative IEC:

$$\frac{dT}{d\bar{x}} = -NTU_2{}^a \begin{bmatrix} \frac{2h^{fin2}}{s^{fin2}} \int_{\bar{z}_2=0}^{\bar{z}_2=1} [T'^{\text{fin2}} - T_2]\, d\bar{Z}_2 + \\ \left(1 - \frac{\partial^{fin2}}{s^{fin2}}\right) (T'^{\text{plate2}} - T_2) \end{bmatrix} (3)$$

The mass conservation equation for water

- For parallel flow IEC:

$$\frac{d\varphi_2}{d\bar{x}} = NTU_2{}^a \left(\frac{1}{Le_2}\right) \begin{bmatrix} \frac{2h^{fin2}}{s^{fin2}} \int_{\bar{z}_2=0}^{\bar{z}_2=1} \tau^{\text{fin2}} [\varphi'^{\text{fin2}} - \varphi_2]\, d\bar{Z}_2 + \\ \left(1 - \frac{\partial^{fin2}}{s^{fin2}}\right) \tau^{\text{plate2}} (\varphi'^{\text{plate2}} - \varphi_2) \end{bmatrix} (4)$$

- For counter flow and regenerative IEC:

$$\frac{d\varphi_2}{d\bar{x}} = -NTU_2{}^a \left(\frac{1}{Le_2}\right) \begin{bmatrix} \frac{2h^{fin2}}{s^{fin2}} \int_{\bar{z}_2=0}^{\bar{z}_2=1} \tau^{\text{fin2}} [\varphi'^{\text{fin2}} - \varphi_2]\, d\bar{Z}_2 + \\ \left(1 - \frac{\partial^{fin2}}{s^{fin2}}\right) \tau^{\text{plate2}} (\varphi'^{\text{plate2}} - \varphi_2) \end{bmatrix} (5)$$

The energy balance equations for fins –

In dry channel of IEC, the energy conservation equation is heat conduction equation considering only sensible heat transfer. The energy balance equation for fins in dry side of IEC:

$$\frac{\partial^2 T^{fin1}}{\partial \bar{Z}_1^2} = -(m^{fin1})^2 (h^{fin1})^2 (T^1 - T^{fin1}) (6)$$

Where,

$$(m^{fin1})^2 = \frac{2\alpha_1}{\lambda^{fin1}\partial^{fin1} + 2(\partial^{fin1})_{foil}(\lambda^{fin1})_{foil}}$$

In wet channel of IEC, the energy conservation equation is combination of both sensible and latent heat transfer on the surface of fin. The energy balance equation for fins in wet channel of IEC:

$$\frac{\partial^2 T^{fin2}}{\partial \bar{Z}_2^2} = -(m^{fin2})^2 (h^{fin2})^2 \begin{bmatrix} (T^2 - T'^{fin1}) - \\ \left(\frac{\tau^{fin2}}{cLe}\right)_2 (\varphi'^{\text{fin2}} - \varphi_2) s_{fin2}^0 \end{bmatrix}$$

(7)

Where,

$$(m^{fin2})^2 = \frac{2\alpha_2}{\lambda^{fin2}\partial^{fin2}}$$

Energy balance equations for air flow-metal surface interface in dry primary air channel:

$$\alpha_1 (T_1 - T'^{\text{plate1}}) \left(1 - \frac{\partial^{fin1}}{s^{fin1}}\right) + \frac{2\alpha_2}{(m^{fin1})^2 s^{fin1} h^{fin1}} \left(\frac{\partial T^{fin1}}{\partial \bar{Z}_1}\right)_{\bar{Z}_1=0} = \left(\frac{\lambda^{plate}}{\partial^{plate}}\right) (T'^{\text{plate2}} - T^{\text{plate1}})$$

(8)

Taking into consideration, the analytical solution of equation (6), the above equation is converted into,

$$\alpha_1(T_1 - T'^{\text{plate1}})\left(1 - \frac{\partial^{fin1}}{s^{fin1}}\right) + \frac{2\alpha_2}{m^{fin1}s^{fin1}}\alpha_1(T_1 - T'^{\text{plate1}})\tanh(h^{fin1}m^{fin1}) = \left(\frac{\lambda^{plate}}{\partial^{plate}}\right)(T'^{\text{plate2}} - T^{\text{plate1}})$$

(9)

Energy balance equations for air flow-metal surface interface in wet working air channel:

$$\left(\frac{\lambda^{plate}}{\partial^{plate}}\right)(T^{\text{plate1}} - T'^{\text{plate2}}) + \alpha_2(T_2 - T'^{\text{plate2}})\left(1 - \frac{\partial^{fin2}}{s^{fin2}}\right) = \left(\frac{\tau^{\text{plate}}\alpha s^0}{cLe}\right)_2(\varphi'^{\text{plate2}} - \varphi_2)\left(1 - \frac{\partial^{fin2}}{s^{fin2}}\right) - \left(\frac{\lambda^{fin2}}{h^{fin2}}\right)\left(\frac{\partial^{fin2}}{s^{fin2}}\right)\left(\frac{\partial T^{fin2}}{\partial \bar{Z}_2}\right)_{\bar{Z}_2=0}$$

(10)

By rearranging the equations (8, 9, 10) we obtain:

$$\left(\frac{P_1}{P_2}\right)NTU_1^a\left[(T_1 - T^{\text{plate1}})\left(1 - \frac{\partial^{fin1}}{s^{fin1}}\right) + \frac{2}{(m^{fin1})^2 s^{fin1}h^{fin1}}\left(\frac{\partial T^{fin1}}{\partial \bar{Z}_1}\right)_{\bar{Z}_1=0}\right] + NTU_2^a(T_2 - T'^{\text{plate2}})\left(1 - \frac{\partial^{fin2}}{s^{fin2}}\right) = NTU_2^a\left(1 - \frac{\partial^{fin2}}{s^{fin2}}\right)\left(\frac{\tau^{\text{plate}}s^0}{cLe}\right)_2(\varphi'^{\text{plate2}} - \varphi_2) - \frac{2}{(m^{fin2})^2 s^{fin2}h^{fin2}}\left(\frac{\partial T^{fin2}}{\partial \bar{Z}_2}\right)_{\bar{Z}_2=0}$$

(11)

The above equation can be converted to final form:

For the parallel flow configuration IEC:

$$\left(\frac{P_1}{P_2}\right)\frac{dT_1}{d\bar{x}} + \frac{dT_2}{d\bar{x}} + \left(\frac{s^0}{c}\right)_2\frac{d\varphi_2}{d\bar{x}} = 0$$

(12)

As the flow of secondary working air is in opposite direction in case of counter and regenerative IEC, the overall balance equation is as follows:

$$\left(\frac{P_1}{P_2}\right)\frac{dT_1}{d\bar{x}} - \frac{dT_2}{d\bar{x}} - \left(\frac{s^0}{c}\right)_2\frac{d\varphi_2}{d\bar{x}} = 0$$

(13)

The above overall balance equations (12,13) are supplemented by the boundary conditions:

1) At the entrance of dry channel, the flow parameters are same for all flow configurations of IEC. They are:

$$T^1 = (T_1^i)_{\bar{x}=0.0,\ \bar{y}=(0.0-1.0)}$$

$$\varphi_1 = (\varphi_1^i)_{\bar{x}=(0.0-1.0),\bar{y}=(0.0-1.0)}$$

2) At the entrance to wet channel, the flow parameters are different for different flow configurations. They are:
   - In parallel flow IEC:

$$T_2 = (T_2^i)_{\bar{x}=0.0,\ \bar{y}=(0.0-1.0)} = (T_1^i)_{\bar{x}=0.0,\ \bar{y}=(0.0-1.0)}$$

$$\varphi_2 = (\varphi_2^i)_{\bar{x}=0.0,\ \bar{y}=(0.0-1.0)} = (\varphi_1^i)_{\bar{x}=0.0,\bar{y}=(0.0-1.0)}$$

   - In counter flow IEC:

$$T_2 = (T_2^i)_{\bar{x}=1.0,\ \bar{y}=(0.0-1.0)} = (T_1^i)_{\bar{x}=1.0,\ \bar{y}=(0.0-1.0)}$$

$$\varphi_2 = (\varphi_2^i)_{\bar{x}=1.0,\ \bar{y}=(0.0-1.0)} = (\varphi_1^i)_{\bar{x}=1.0,\bar{y}=(0.0-1.0)}$$

   - In regenerative IEC:

$$T_2 = (T_2^i)_{\bar{x}=1.0,\ \bar{y}=(0.0-1.0)} = (T_0^i)_{\bar{x}=1.0,\ \bar{y}=(0.0-1.0)}$$

$$\varphi_2 = (\varphi_2^i)_{\bar{x}=1.0,\ \bar{y}=(0.0-1.0)} = (\varphi_1^i)_{\bar{x}=1.0,\bar{y}=(0.0-1.0)} = (\varphi_0^i)_{\bar{x}=1.0,\bar{y}=(0.0-1.0)}$$

The relation between water vapor's temperature and its pressure is given by,

$$p^{sat} = 6.107(0.0726T - 2.912 \times 10^{-4}T^2 + 8.33 \times 10^{-7}T^3) \quad (14)$$

The humidity ratio can be calculated by using equation,

$$\varphi^{sat} = \frac{p^{sat}}{p^a - p^{sat}} \quad \text{Kg of moisture/Kg of dry air}$$

III. CONCLUSION

A numerical study describing three different indirect evaporativeaircoolers (parallel-flow, counter-flow, and regenerative)were presented in this paper. A mathematical model of IEC is developed by using heat and mass balance equations, inlet and exit boundary conditions and empirical relations. The mathematical model thus developed can be utilized to predict the performance and process parameters of an experimental model.

NOMENCLATURE:

cspecific heat capacity(J/Kg.K)
s^0 specific heat of evaporation of water
T Temperature(°C).
P heat capacity rate of fluid (W/K).
pvapor partial pressure (Pa).
p^aatmospheric pressure (Pa).
sfin pitch(m).
m water vapor mass transfer rate (kg/s).
llength of cooler along the stream (m).

Special symbols

φ humidity ratio (Kg/Kg).
$\bar{x}\bar{x} = X/l_x$ -relative X coordinate.
$\bar{y}\bar{y} = Y/l_y$- relative Y coordinate.
∂ thickness(mm).
λ thermal conductivity(W/mK).
τsurface wettability factor(τ∈ (0,1)).
Le Lewis factor.
St Stanton number.
Nu Nusselt number.
NTU Number of transfer units.

Subscripts and Superscripts:

plate channel plate.
finfin surface.
1 main air stream.
2 secondary (working) air stream.
a referenced to metal plate surface
b conditions at air –water interface T.
heatheat transfer.
massmass transfer.
iinlet.
o outlet.

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