Numerical Investigation of a Novel Plate-Fin Indirect Evaporative Cooling System Considering Condensation

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Abstract: An indirect evaporative cooling system combining with thermoelectric cooling technology (i.e., TIEC system) is proposed, in which a counter-flow plate-fin indirect evaporative cooler is inserted with thermoelectric cooling (i.e., TEC) modules. In hot and humid climate, condensation may occur on the dry channel surface of the cooler. For the TIEC system, with the aid of TEC technology, the surface temperature of the dry channel can be much lower than that of a traditional indirect evaporative cooler, thus, the condensation from the primary air is more likely to take place. A numerical model of this novel TIEC system is developed with specifically taking condensation from primary air into account. Detailed performance analysis of the TIEC system is carried out. Analytical results found that the condensation from primary air reduces the dew point effectiveness by up to 45.0% by weakening the sensible heat transfer but increases the coefficient of performance by up to 62.2% by increasing the latent heat transfer, under given conditions. The effects of main operating conditions, such as the electrical current I and number n of TEC modules, inlet temperature $T_{p,i}$, humidity ratio $RH_p$ and velocity $V_p$ of the primary air, and the mass flow rate ratio $x$ of secondary to primary air, are investigated under non-condensation and condensation states. It is shown that condensate is more easily produced under higher $I$, $n$, $T_{p,i}$, $RH_p$, $x$ and lower $V_p$.

Keywords: thermoelectric cooling; indirect evaporative cooling; condensation; heat transfer; mass transfer

1. Introduction

Evaporative cooling technology, which uses latent heat of water evaporation as the cooling source, has aroused wide attention for the last few years in building energy conservation due to its unique advantages of energy saving, environmental friendliness, simple structural configuration and easy maintenance [1]. Unlike direct evaporative coolers (i.e., DEC) where product air is in direct contact with water, indirect evaporative coolers use paired dry and wet channels. In the wet channel, water evaporates into the working air, and in the adjacent dry channel, product air is cooled down without the increase of humidity [2]. Indirect evaporative cooling system (i.e., IEC system) is suitable for many applications in public buildings and also in agriculture and industrial area [3]. For the IEC system, the main constraint of limiting its wide use is the wet bulb temperature of ambient air. In hot and humid conditions, high wet-bulb temperature of ambient air limits the supply air temperature, which restricts the application of indirect evaporative coolers. Therefore, new technologies and methods are needed for overcoming the shortcoming of the indirect evaporative cooler and improve its application potential.

Many researchers have proposed novel cooler configurations to further enhance the performance of IEC systems, and achieve sub-wet bulb temperature. Hasan [3] proposed an idea to cool the product air to the sub-wet bulb temperature by branching the working air from the product air. In this method, the product was indirectly pre-cooled before it was
finally cooled. Anisimov and Pandelidis [4] numerically analyzed the advantages and disadvantages of cross-flow, counter-flow, regenerative and parallel-flow IEC heat exchangers. Multistage IEC systems and various hybrid systems of combining IEC and other cooling technologies have also been studied. Cooling performance of a two stage IEC/DEC system was experimentally investigated in various climatic conditions by Heidarinejad et al. [5]. The effectiveness of IEC/DEC system ranged from 108% to 111% while the effectiveness of IEC stage varied from of 55% to 61%. Moshari et al. [6] put forward three type two stage IEC/IEC systems, and found that the wet-bulb effectiveness of the three systems were all obviously higher than that of one stage IEC system. Khalajzadeh et al. [7] presented a hybrid system which used a ground-coupled circuit to precool the entering air of an IEC. The simulation results found that the hybrid system could cool the air to below the wet-bulb temperature. Farahani et al. [8] studied a two-stage nocturnal radiative/IEC system for conditions in Tehran. The results showed that by using the first stage nocturnal radiative precooling system, the system effectiveness could considerably increase. The authors [9] have theoretically investigated the performance of a hybrid system combining TEC and plate-type indirect evaporative cooler (i.e., TIEC system). It was shown that the TIEC system could cool the air to below the dew point temperature, and meanwhile maintain high COP by choosing appropriate working parameters of thermoelectric modules.

Simulation models, which consider different factors such as heat conduction, evaporative water, variable Lewis factor and loss water temperature variation, have been proposed and used for parameter analysis, performance prediction and configuration optimization of indirect evaporative coolers or hybrid cooling systems with indirect evaporative coolers [10–12]. Most of the models only consider the sensible heat transfer from the primary air. Nevertheless, condensation probably takes place when the dew point temperature of ambient air is high, which is gradually obtaining attention in last few years. Some researchers conducted studies to investigate and prove the impact of the condensation from primary air on performance of IEC systems [13–15]. For TIEC system as mentioned above [9], the surface temperature of the product air channel would be much lower than that of the traditional indirect evaporative cooler due to the assistance of TEC modules, thus, condensation from the primary air is more likely to occur. Thus, to better predict the performance of the TIEC system, a model including the influence of the condensation from the primary air is needed.

To enrich the previous research, in this paper, an analytical model for TIEC system especially considering condensation from primary air is developed. Additionally, a plate fin heat exchanger instead of a plate heat exchanger as in the previous study is adopted for the new TIEC system to further enhance the heat and mass transfer. The impact of the condensate from the primary air on the TIEC system performance will be specifically analyzed under various operating parameters, including the electrical current and number of TEC modules, inlet temperature, humidity ratio, and velocity of the primary air, and also the mass flow rate ratio of secondary air to primary air.

2. System Description

Figure 1a displays a schematic diagram of the novel counter-flow plate-fin IEC system with TEC modules. In the dry channel, Primary air flows downward. At the dry channel outlet, primary air is partially used as supply air, and the rest is redirected into the wet channel from bottom up as the secondary working air. The water distributed on the wet channel surface partially evaporates into the secondary air as it flows downward due to gravity. Figure 1b illustrates the structure of a counter-flow plain-fin indirect evaporative cooler inserted with TEC modules. Plain fins are arranged in both wet and dry channels. Sandwiched between dry and wet channels, TEC modules are installed in a way that the hot side is connected to the wet channel and the cold side to the dry channel. Dark blue arrow shows the flow direction of the primary air, green arrow of the secondary air, and light blue arrow of the water.
3. Mathematical Model

Figure 2 demonstrates a control volume of the counter-flow plate-fin TEC/indirect evaporative cooler, including half wet channel, the partition plate with TEC modules, and half dry channel. The control volume finite difference method is used to build a steady-state numerical model for the heat and mass transfer processes in the cooler based on the basic thermoelectric cooling theory and heat and mass transfer laws. The numerical model of the proposed TEC/indirect evaporative cooler just combines the classic models of the thermoelectric cooling module and the indirect evaporative cooler, which are well recognized and extensively used in open literature [10,16]. Along the $z$-axis, the cooler is discretized into two hundred segments, and each segment has a length of $d_z$. Condensation water may appear on the dry channel surface. Several common assumptions are made to simplify the model referring to Reference [9].

Figure 1. (a) Schematic diagram of a counter-flow plate-fin TIEC system; (b) a counter-flow plate-fin indirect evaporative cooler with TEC modules.

Figure 2. Control volume of the counter-flow plate-fin TEC/indirect evaporative cooler.
For TEC modules, the cooling capacity $Q_c$ in the cold side, and the heat released to the hot side $Q_h$ are expressed as [16],

$$Q_c = n \left[ a_0I_T - K_0(T_h - T_c) - \frac{1}{2} R_0 I^2 \right]$$  \hspace{1cm} (1)

$$Q_h = n \left[ a_0I_T - K_0(T_h - T_c) + \frac{1}{2} R_0 I^2 \right]$$  \hspace{1cm} (2)

where $T_h$ and $T_c$ are the hot and cold junction temperatures of TEC modules. $n$ is TEC module number, and $I$ is the electrical current.

The heat transfer rate in each segment with height $dz$ between TEC module hot junction and the water, $dQ_h$, is given by,

$$dQ_h = \frac{Q_h}{H} dz = K_h(T_h - T_w)dA$$ \hspace{1cm} (3)

where $K_h$ is the overall thermal conductance from TEC module hot junction to the water.

In each segment, heat balance of the wet channel is expressed as follows,

$$dQ_h = dQ_a + dQ_w$$ \hspace{1cm} (4)

$$dQ_a = dQ_{a,s} + dQ_{a,l}$$ \hspace{1cm} (5)

where $dQ_w$ is heat increment of water, and $dQ_a$ is the heat transfer rate on the secondary air/water interface, including latent and sensible heat transfer rate (i.e., $dQ_{a,l}$ and $dQ_{a,s}$). $dQ_w$, $dQ_{a,s}$ and $dQ_{a,l}$ are calculated by using Equations (6)–(8),

$$dQ_w = (m_w + \frac{\partial m_w}{\partial z}dz)c_w(T_w + \frac{\partial T_w}{\partial z}dz) - m_w c_w T_w$$ \hspace{1cm} (6)

$$dQ_{a,s} = h_a(T_w - T_a)\eta_{o,a}dA$$ \hspace{1cm} (7)

$$dQ_{a,l} = (r_w k_d)p(W_w - W_a)\beta dA$$ \hspace{1cm} (8)

where $h_a$ and $k_d$ are heat and mass transfer coefficients of secondary air, $\eta_{o,a}$ is overall fin efficiency of the secondary air channel. $r_w$ is the specific enthalpy of water vapor at local temperature and $\beta$ is the surface wettability factor.

The mass exchange between secondary air and water can be given by,

$$\frac{\partial m_w}{\partial z}dz = m_a \frac{\partial W_a}{\partial z}dz = k_{d,a}(W_w - W_a)\beta dA$$ \hspace{1cm} (9)

When local plate surface temperature $T_{wall}$ is higher than the dew point temperature of primary air $T_{p,dp}$, primary air only transfers sensible heat $dQ_{p,s}$ to cold side of TEC modules. The heat balance in the dry channel within each segment can be written as,

$$dQ_c = dQ_{p,s}$$ \hspace{1cm} (10)

$$dQ_{p,s} = h_p(T_p - T_{wall})\eta_{o,p}dA$$ \hspace{1cm} (11)

where $T_p$ is the primary air temperature, $\eta_{o,p}$ is overall fin efficiency of the primary air channel.

When $T_{wall}$ is lower than $T_{p,dp}$, both sensible and latent heat transfer occur. Then, we have,

$$dQ_c = dQ_{p,s} + dQ_{p,l}$$ \hspace{1cm} (12)

$$dQ_{p,l} = (r_w k_d)p(W_p - W_{wall})\beta dA$$ \hspace{1cm} (13)
The mass exchange between primary air and wall is as follows,

\[ m_p \frac{\partial W_p}{\partial z} dz = - \frac{\partial m_{\text{cond}}}{\partial z} dz = k_{d,p} (W_p - W_{\text{wall}}) \beta dA \]  

(14)

The heat transfer coefficient \( h_a \) and \( h_p \) can be calculated by [17],

\[ h = 36.31 (\rho u)^{0.68} \left( \frac{L}{D_e} \right)^{-0.08} \]  

(15)

where \( \mu \) is the air velocity, \( D_e \) is the channel equivalent diameter, and \( L \) is the air channel length. Mass transfer coefficient \( k_{d,a} \) and \( k_{d,p} \) can be calculated by,

\[ k_d = \frac{h}{c_p} \]  

(16)

The heat transfer coefficient of water film is written as [17],

\[ Nu_w = \frac{h_w \delta_c}{\lambda_w} = 1.88 \]  

(17)

where \( \delta_w \) is the water film thickness calculated by using Equation (18) as follows,

\[ \delta_w = \left( \frac{3v_w m_w}{\rho_w g L} \right)^{\frac{1}{3}} \]  

(18)

Normally, dew-point effectiveness \( \varepsilon_{dp} \) and coefficient of performance (COP) are used to evaluate the overall system performance, which can be obtained by using Equations (18) and (19) [18],

\[ \varepsilon_{dp} = \frac{T_{p,i} - T_{p,o}}{T_{p,i} - T_{p,dp}} \]  

(19)

\[ \text{COP} = \frac{Q_c}{N} \]  

(20)

where \( T_{p,i} \) and \( T_{p,o} \) are inlet and outlet temperature of primary air. Total energy consumption of pumps, fans, and TEC modules \( N \) is calculated by the following equation [19],

\[ N = N_p + N_s + N_e = \frac{G_p \Delta P_p}{\eta_p} + \frac{G_s \Delta P_s}{\eta_s} + n [I^2 R_0 + a_0 I (T_h - T_c)] \]  

(21)

Fan efficiencies \( \eta_p \) and \( \eta_s \) are assumed to be 0.75 [20].

When condensation occurs in the dry channel, some new parameters need to be introduced to indicate the effect of latent heat exchange, i.e., enlargement ratio \( \xi \) and dehumidification ratio \( \varphi \),

\[ \xi = \frac{Q_{p,l} + Q_{p,s}}{Q_{p,s}} \]  

(22)

\[ \varphi = \frac{W_{p,l} - W_{p,o}}{W_{p,i}} \]  

(23)

4. Results and Discussion

Numerical investigations have been conducted to evaluate the performance of the novel TIEC system under different states (i.e., condensation state and non-condensation state). Table 1 shows key geometrical and operating parameters. Commonly used commercial Bi₂Te₃ based TEC modules (CP1.4-127-06L) are used as thermoelectric materials, and the corresponding parameters are chosen according to Reference [9].
Table 1. Key geometrical and operating parameters.

| Parameter                      | Value       |
|--------------------------------|-------------|
| Primary air dry-bulb temperature | 35 °C       |
| Primary air velocity           | 3 m/s       |
| Primary air relative humidity  | 30–70%      |
| Mass flow rate ratio $x_{r,p}$ (i.e., $m_r/m_p$) | 0.42        |
| Channel length                 | 0.5 m       |
| Channel height                 | 0.5 m       |
| Channel width                  | 5 mm        |
| Fin thickness                  | 0.2 mm      |
| Fin pitch                      | 10 mm       |
| Intermediate plate thickness   | 1.5 mm      |

Figure 3a shows the variation trends of COP and $\varepsilon_{dp}$ with the electric current $I$ under non-condensation (i.e., Non-cond) and condensation (i.e., Cond) conditions. It can be seen that higher $I$ leads to higher $\varepsilon_{dp}$, but lower COP. When the RH$_p$ keeps as 30%, condensation occurs in dry channel when the current $I$ is higher than 1.3 A. When the RH$_p$ is 50%, condensation occurs when the current $I$ is higher than 0.6 A. And when the RH$_p$ is 70%, the condensation occurs when the current $I$ is lower than 0.5 A. To conclude, when the RH$_p$ is higher, condensation is more likely to occur under lower $I$. As is known that condensation occurs when the plate surface temperature of the primary air channel is decreased to lower than dew point temperature $T_{p,dp}$. And when the inlet primary air temperature is constant, higher RH$_p$ leads to higher $T_{p,dp}$. Thus, the plate surface temperature leading to condensation could be reached under lower $I$. Under both non-condensation and condensation state, $\varepsilon_{dp}$ increases with the RH$_p$ increasing from 30% to 70%. However, with the increase of RH$_p$, COP under non-condensation state decreases and COP under condensation state increases. In addition, condensation from primary air would decrease $\varepsilon_{dp}$ up to 45.0% and increase the COP up to 62.2% by comparing the red lines under condensation conditions and black lines under non-condensation conditions. Once condensation occurs, latent heat transfer would appear and release heat to the plate, resulting in the increase of plate temperature, and then the increase of the outlet air temperature $T_{p,o}$, which finally leads to the decrease of $\varepsilon_{dp}$ ($\varepsilon_{dp} = (T_{p,i} - T_{p,o})/ (T_{p,i} - T_{p,d})$). As shown in Figure 1b, when condensation occurs, the sensible heat transfer rate $Q_s$ decreases, but the latent heat transfer rate $Q_l$ increases, thus, total heat transfer rate $Q_c$ increases, which finally raises the value of COP (COP = $Q_c/N$). Therefore, the latent heat transfer $Q_l$ plays a significant role in the heat and mass transfer process under condensation conditions. Moreover, when $I$ is higher, the $\varepsilon_{dp}$ under condensation state is much lower than those under non-condensation state. That is because that when $I$ is higher, the plate temperature is lower, the driving force of mass transfer is larger, and thus, more latent heat is released. Figure 3c further shows that with considering condensation, both the enlargement ratio $\xi$ and dehumidification ratio $\phi$ rise with the increases of $I$ and RH$_p$. The value of $\xi$ could even reach 2.73 when $I$ is 4 A and RH$_p$ is 70%, which means the $Q_l$ from condensation is much more higher than $Q_s$. The dehumidification ratio $\phi$ increases by more than 50% when $I$ is 4 A and the RH$_p$ is 70%.

Figure 4 shows the variation trends of COP and $\varepsilon_{dp}$ with the TEC module number $n$ under different RH$_p$. It is shown that with the increase of $n$, COP decrease, while $\varepsilon_{dp}$ increases. When RH$_p$ is higher, condensation occurs under lower $n$. With the RH$_p$ as 30%, non-condensation state takes place with $n$ ranging from 8 to 25. With the RH$_p$ as 50%, $\varepsilon_{dp}$ decreases by up to 7.5% and COP increases by up to 16.5% once condensation occurs when TEC module $n$ is higher than 12. As RH$_p$ increases to 70%, condensation occurs throughout the range of TEC module $n$, which leads to a remarkable decrease of $\varepsilon_{dp}$ by up to 26.1% and increase of COP by up to 58.8%.
the enlargement ratio $\xi$ and dehumidification ratio $\phi$ rise with the increases of $I$ and $p_{RH}$. The value of $\xi$ could even reach 2.73 when $I$ is 4A and $p_{RH}$ is 70%, which means the $Q_f$ from condensation is much higher than $Q_s$. The dehumidification ratio $\phi$ increases by more than 50% when $I$ is 4A and the $p_{RH}$ is 70%.

Figure 3. (a) COP and $\varepsilon_{dp}$ versus $I$; (b) $Q_s$ and $Q_l$ versus $I$; (c) Enlargement ratio $\xi$ and dehumidification ratio $\phi$ versus $I$. 


Figure 4. COP and $\epsilon_{dp}$ versus $n$ under various $RH_p$.

Figure 5 displays the value of COP and $\epsilon_{dp}$ with primary air inlet temperature $T_{p,i}$ under non-condensation and condensation states. With $n$ as 10 and $I$ as 0.5 A, condensation occurs when $T_{p,i}$ is higher than 39 °C. With $n$ as 15 and $I$ as 0.5 A, condensation occurs when $T_{p,i}$ is higher than 29 °C. With $n$ as 10 and $I$ as 1.0 A, condensation occurs when $T_{p,i}$ is lower than 25 °C. To conclude, when $n$ and $I$ are higher, condensation tends to occur under lower $T_{p,i}$. When the $T_{p,i}$ increases from 25 °C to 45 °C, COP increases monotonously under all given conditions. The dew point effectiveness $\epsilon_{dp}$ increases with the $T_{p,i}$ growing under non-condensation state. However, under condensation state, there exists a maximum $\epsilon_{dp}$ of 0.96 when $n$ is 10 and $I$ is 0.5 A, a maximum $\epsilon_{dp}$ of 1.01 when $n$ is 15 and $I$ is 0.5, and a maximum $\epsilon_{dp}$ of 1.16 when $n$ is 10 and $I$ is 1.0 A. In addition, in the given ranges of $T_{p,i}$, condensation from primary air would decrease $\epsilon_{dp}$ by up to 24.5% and increase the COP by up to 31.2% by comparing the red lines under condensation conditions and black lines under non-condensation conditions.

Figure 5. COP and $\epsilon_{dp}$ versus $T_{p,i}$ under various $I$ and $n$.

Figure 6 presents the impact of inlet velocity of primary air $V_p$ on the COP and $\epsilon_{dp}$. COP increases and $\epsilon_{dp}$ decreases with $V_p$ under given condition. The increase of $V_p$ could lead to a larger mass flow rate and a larger cooling capacity, resulting in the increase of COP. However, the increase of $V_p$ weakens the heat transfer between plate and primary air, leading to the decrease of the primary air outlet temperature, and then the decrease of $\epsilon_{dp}$. The condensation disappears when $V_p$ is larger than 2.5 m/s when $RH_p$ is 50%. And when
RH_p is 70%, the gape between ε_dp curves under non-condensation and condensation states becomes more narrow with the increase of V_p. The above results show that condensation is more likely occur under lower V_p. In addition, in the given rages of V_p, condensation from primary air would decrease ε_dp by up to 25.8% and increase the COP by up to 49.8%.

Figure 6. COP and ε_dp versus V_p under various l and n.

Figure 7 illustrates variation trends of COP and ε_dp with mass flow rate ratio x of secondary air to primary air. It can be seen that with RH_p as 70%, condensation occurs throughout the given range of x, with RH_p as 50%, condensation occurs when x is greater than 0.5, and with RH_p as 30% non-condensation occurs. To conclude, when RH_p is higher, condensation is more likely to occur under lower x. ε_dp increases monotonously with x under certain RH_p. In the given ranges of x, condensation from primary air would decrease ε_dp up to 26.6% and increase the COP up to 36.9% by comparing the red lines under condensation conditions and black lines under non-condensation conditions. Moreover, a maximal COP exists with an optimal x.

Figure 7. COP and ε_dp versus mass flow rate ratio x under various l and n.
5. Conclusions

A numerical model of a novel thermoelectric assisted plate-fin indirect evaporative cooling system with especially considering effect of condensation from primary air is developed to evaluate the system performance. With the assistance of TEC modules, the plate surface temperature of the dry channel could be much lower than the dew point temperature of the fresh air, thus, condensate easily occurs. The condensation of the primary air reduces the dew point effectiveness by up to 45.0% by weakening the sensible heat transfer and raises the COP by up to 62.2% by increasing the latent heat transfer under given conditions. Thus, condensation from the primary air is nonnegligible for the performance analysis of the novel cooling system. When $\text{RH}_p$ is higher, condensation is more likely to occur under lower $I$, $n$ and $x$. Higher $I$ and higher $n$ lead to lower plate temperature, which promotes the condensation. When $n$ and $I$ are higher, condensation tends to occur under lower $T_{p,i}$. In addition, condensation is more likely occur under lower $V_p$.

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Abbreviations

| Symbol | Description |
|--------|-------------|
| $A$    | Area ($m^2$) |
| $c$    | Specific heat ($J \cdot kg^{-1} \cdot K^{-1}$) |
| COP    | Coefficient of performance |
| $H$    | Height ($m$) |
| $h$    | Heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$) |
| $I$    | Working current ($A$) |
| $k_d$  | Mass transfer coefficient ($kg \cdot m^{-2} \cdot s^{-1}$) |
| $K_0$  | Total thermal conductance of TEC module ($W \cdot K^{-1}$) |
| $m$    | Mass flow rate ($kg \cdot s^{-1}$) |
| $N$    | Energy consumption ($W$) |
| $n$    | TEC module number |
| $\Delta P$ | Pressure drop ($Pa$) |
| $Q$    | Heat transfer rate ($W$) |
| $R_0$  | Total electrical resistance ($\Omega$) |
| RH     | Relative humidity |
| $r_w$  | Specific enthalpy of water vapor ($J \cdot kg^{-1}$) |
| $T$    | Temperature ($^\circ C$) |
| $W$    | Humidity ratio ($kg \cdot kg^{-1}$) |

Greek symbols

| Symbol | Description |
|--------|-------------|
| $\alpha_0$ | Seebeck coefficient ($V \cdot K^{-1}$) |
| $\beta$  | Surface wettability factor |
| $\epsilon_{dp}$ | Dew point effectiveness |
| $\eta_0$ | Overall fin efficiency |
| $\delta$ | Thickness ($m$) |
| $\sigma$ | Condensation ratio |
| $\xi$   | Enlargement ratio |
| $\varphi$ | Dehumidification ratio |
Subscripts
a Secondary air
c Cold-side
cond Condensation
h Hot-side
i Inlet
l Latent
o Outlet
p Primary air
dp Dew point
s Sensible
w Water film

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