Accepted Manuscript

Approaches to energy efficiency in air conditioning: A comparative study on purge configurations for indirect evaporative cooling

Seung Jin Oh, Muhammad Wakil Shahzad, Muhammad Burhan, Wongee Chun, Chua Kian Jon, M. KumJa, Kim Choon Ng

PII: S0360-5442(18)32288-6

DOI: <https://doi.org/10.1016/j.energy.2018.11.077>

Reference: EGY 14175

To appear in: *Energy*

Received Date: 31 July 2018

Revised Date: 11 November 2018

Accepted Date: 20 November 2018

Please cite this article as: Oh SJ, Shahzad MW, Burhan M, Chun W, Jon CK, KumJa M, Ng KC, Approaches to energy efficiency in air conditioning: A comparative study on purge configurations for indirect evaporative cooling, *Energy* (2018), doi: <https://doi.org/10.1016/j.energy.2018.11.077>.

This is a PDF file of an unedited manuscript that has been accepted for publication. As a service to our customers we are providing this early version of the manuscript. The manuscript will undergo copyediting, typesetting, and review of the resulting proof before it is published in its final form. Please note that during the production process errors may be discovered which could affect the content, and all legal disclaimers that apply to the journal pertain.

Approaches to Energy Efficiency in Air Conditioning: A Comparative Study on Purge Configurations for Indirect Evaporative Cooling

Seung Jin Oh¹, Muhammad Wakil Shahzad¹, Muhammad Burhan¹, Wongee Chun², Chua Kian Jon³, M KumJa³, Kim Choon Ng^{1,*}

¹Water Desalination and Reuse Center, King Abdullah University of Science & Technology, Thuwal 23955-6900, Kingdom of Saudi

²Department of Nuclear and Energy Engineering, Jeju National University, 1 Ara 1 dong, Jeju 63243, Korea

³Department of Mechanical Engineering, National University of Singapore, Engineering Drive 1, Singapore 117575

Abstract

This paper presents a mathematical model to enable a more in-depth understanding of the combined heat and mass transfer processes of a counter-flow indirect evaporative cooler operating in semi-arid regions. A series of simulation was conducted to compare the performances between a single-purge and four-purge configurations. Furthermore, the effect of several parameters on the performance of the single-purge configuration was investigated under the baseline condition. The developed model was validated against experimental data from literatures to within 2% discrepancy. Key results revealed that the single-purge configuration produced higher cooling capacity (around 20% higher), as well as higher dew-point effectiveness under the same operating conditions. A parametric study was also made on the single-purge configuration. The results revealed that lower product air temperature and higher dew-point effectiveness were achieved with longer channel length and larger purge ratio. However, the maximum cooling capacity approached its maximum with longer channel length and 35% purge ratio.

Keywords Indirect evaporative cooler, Dew-point evaporative cooler, Cooling capacity, Purge configuration, Numerical simulation

1. Introduction

The global air-conditioned (AC) and refrigeration industry is valued at US\$150 billion by 2018 and is expected to reach US\$ 230 billion by the end of 2028 with a CAGR(compound annual growth rate) of 4.1% [1]. Although AC in buildings has improved human's lifestyle and work efficiency in commercial buildings, it is inevitably accompanied with the negative effects such as the emissions of greenhouse gases (GHG) directly via refrigerant emissions and indirectly through electricity generation by burning fossil fuels [2–4]. Moreover, the global energy demand for space cooling is expected to triple by 2050 [4]. Of the 2.8 billion people living in the hotter regions of the world such as KSA, only 8% currently possess AC, and as incomes and living standards improve in many such countries, the demand for AC is expected to rise sharply. As a consequence, AC usage will be the second-largest consumer of global electricity after an industrial sector by 2050 [4–7].

* Corresponding author: kimchoon.ng@kaust.edu.sa

Over the past decades, a hot and dry climate country, Saudi Arabia (except coastal cities along Red Sea and Gulf) has spent more than 50% of the annual electricity production (272 TWh, 2014) for heating, ventilation and air conditioning (HVAC) purposes, amounting to 139 TWh per annum [8]. Should this cooling energy demand is unabated, the future sustainability outlook in energy and environment efficacy of the mentioned country may be untenable [9–11]. In conventional air-conditioning systems, the compressor efficiency has shown a significant improvement chronologically from 1.2 kW/RT in the 1990s to 0.85 ± 0.05 kW/RT. Since 2000, however, this improvement trend has leveled asymptotically [12–14], implying that efficiency improvement for compressor stages and heat exchangers have been saturated.

Although the indirect evaporative cooling (IEC) process is not new technology, it can be an out-of-box solution to the aforementioned challenge especially in a desert climate condition where air-humidity level maintains below 10g/kg throughout the year. Conventional indirect evaporative cooler (IEC) is based on heat and mass transfer between two separated streams of air that flow in a dry channel and a wet channel [15]. However, the conventional IEC has the limit that the temperature of product air cannot go below the wet-bulb temperature of the working air. For such reasons, IEC has been further improved based on a M-cycle conception [16], which also called a dew-point evaporative cooler and regenerative evaporative cooler (REC) [17,18]. In order to control indoor humidity level, IEC can be also integrated with a heat-driven dehumidification system such as a liquid desiccant dehumidifier and a solid desiccant dehumidifier [19–21]. Alternatively, it can be combined with a direct evaporative cooler [22]. Other than air-conditioning application, IEC has been also applied to a power plant [23] and a mine refuge chamber [24]. The key features of an IEC are that it employs only water instead of a refrigerant and its operation is predicated on the design of a generic wet-and-dry channels where the heat and mass transfer processes are imposed onto the product and purged air.

Many experimental and simulation works have been performed with various configurations of IEC under various operating conditions. Xu et al. [25] developed a novel complex heat and mass exchanger (HMX) for higher performance of a dew-point evaporative cooler. Authors removed supporting guides inside channels and employed a corrugated surface for larger heat and mass transfer area. Dew-point effectiveness was found to be in the range 67-76.3% and the wet-bulb effectiveness in the range 100-109.8%. Duan et al. [26] investigated experimentally the operational performance of a regenerative evaporative cooler under various conditions by analyzing the wet-bulb effectiveness and EER as well as cooling capacity. The results showed that wet-bulb effectiveness and EER increased by 31% and 40%, respectively compared conventional indirect cooler. Kim et al. [27] investigated the performance of IEC with two different cross-flow configurations. They found that the mass flow rate of the working air is the key to enhance the performance of the IEC. Hasan [28] proposed an analytical model of IEC with a modified e-NTU method that can be applicable to any type of IEC namely a regenerative, counter-flow and parallel-flow cooler. They redefined heat capacity rate parameters and heat transfer coefficient. Dizaji et al. [29] also developed an analytical model for multi-stage M-cycle exchangers that has perforated exchangers. Their results have been compared with the numerical solution and experimental results obtained from literatures and have found to be in good agreement. Lin et al. [30] simulated the transient and steady-state performance of the dew-point evaporative cooler using a 2-D mathematical model, and compared the results with experimental results. They also investigated the effect of water spray on a channel plate temperature profiles in both horizontal and vertical plate orientations. The governing equations include momentum, continuity, energy balance and species balance. They also concluded that a horizontal

orientation of the channel plate is more efficient than a vertical orientation as it exhibited smaller temperature variation and faster response. Jafaian et al. [31] employed a realistic boundary condition to develop a more accurate numerical model of a dew-point evaporative cooler. Thermal boundary condition on the separating wall was neither constant surface temperature nor constant surface heat flux. They also compared the results of 2D and 3D models and concluded that the 2 D model is a favorable option since the maximum errors between two models were less than 4.5%. Comino et al. [32] presented a simplified model of IEC based on experimental results. Authors compared the results of the simplified model with those of a detailed model and experimental data.

Many attempts have been conducted for the optimization of IEC. Balyani et al. [33] analyzed economic and environmental as well as thermal comfort analysis of IEC for selecting the best cooling technology for various climates. Zhao et al.[34]investigated the feasibility of a polygonal-sheets-stacked heat and mass exchanger for air conditioning in buildings at seven cities of China. Duan et al. [35]investigated experimentally the energy saving potential of a prototype of a counter-flow regenerative cooler when it applied to China's various regions. Jafarian et al. [35] developed GMDH-type neural networks for modeling and optimization of a flat plate dew-point counter flow IEC. Their results indicated that the COP was improved by 36.6% with hot and dry climate. Sohani et al. [36] conducted a multi-objective optimization to find the best design of two different dew-point evaporative coolers(i.e., counter-flow and cross- flow configurations). It was found that the counter-flow configuration was the ideal option in very hot and dry regions (Riyadh), while a cross-flow configuration was better in other investigated climates(Ahmedabad, London, Windsor). Authors also implemented GMDG model for optimization of COP and cooling capacity with 12 different climates [37]. Bi et al. [38] concluded that an optimum management system of IEC could achieve 87.7~91.6% electricity consumption saving in Data Centers in 10 typical cities under 10 climatic conditions.

National Renewable Energy Laboratory (NREL) evaluated three multistage IECs through observation, measurement, and verification of the performance [39]. However, the studied multistage IECs were based on a cross-flow configuration. Thus far, little studies have been conducted on the performance of a counter-flow multi-purge configuration of IEC. In this paper, a mathematical model is established in order to conduct a series of numerical simulation for a counter-flow generic cell of IEC with two purge configurations namely, single purge and four purges. From the simulation results, comparative analyses are carried out in order to identify the differences. Further analyses are made to investigate the performance of a generic cell with the single-purge configuration by examining the effect of some important geometrical and operating parameters.

2. Concept of Indirect Evaporative Cooler

The IEC differs from the conventional adiabatic cooling or commonly known as the cooling towers, swarm coolers, etc., where the air stream experience changes in both the temperature and absolute humidity. For the IEC, the primary air is flowing in a dry channel whilst a small fraction of the product air is purged into a wet channel, where the latter picks up the water vapor from the dedicated "hydrophilic membranes" that separates the evaporative moist air flowing in a counter-flow direction. Thermodynamically, it incurs two major energy losses to achieve cooling of the primary air stream: Firstly, energy is imparted to the air by a fan for maintaining the necessary air flow in all channels and secondly, it continuously purges a fraction of the conditioned air of the non-wetted channel so as to re-initiate the evaporative cooling in successive wetted-channels.

Fig.1 (a) depicts the single-purge configuration where the air is diverted at the exit of dry channel and diverted to the wet channel with a flow rate of 30~40% of the inlet air. Fig. 1 (b), on the other hand, shows the four-purge configuration that injects part of the product air into the wet channel. The four-purge configuration is further classified into two types. For the first type (type A), the air flowing the dry channel is purged from each section of the dry channel and then added to a preceding working air stream. For instance, the first purged air from section 1 is mixed with the working air flowing from section 2. Subsequently, the mixed working air stream with lowered humidity ratio is then supplied to the next the section of the wet channel while increasing evaporation potential. For the other type (type B), the air is purged from the end of the dry channel likewise the single-purge configuration. However, the purged air is flowing through another small passage from which it is purged into each section of the wet channel with the same temperature and humidity ratio of the product air. In this case, both temperature and humidity ratio of the mixed air at each section of the wet channel are lowered. It is noteworthy that the IECs studied in this study are based on a dew-point evaporative cooler (i.e., M-cycle).

Fig. 1 Schematic of the proposed IECs: (a) single-purge configuration, (b) four-purge configuration with two purge methods.

3. Mathematical model for IEC

The mathematical model was established to observe the performance of the counter-flow indirect evaporative cooler (IEC) with two purge configurations. The differential control volume consists of half the height of the dry and wet channels, the separating plate, and the water film. To simplify the heat and mass transfer processes, the following assumptions were made:

- (1) The air streams in both the dry and wet channels are at steady state for the heat and mass transfer analysis.

- (2) Air flow is laminar and fully developed, and heat and mass transfer coefficients are constant.
- (3) Heat losses to the surroundings are negligible (adiabatic condition).
- (4) The wet surfaces are saturated locally with the water film.
- (5) The air velocity and temperature distributions are 1-D with respect to its axial direction.

The dry air channel involves sensible cooling by forced convective heat transfer, leading the change of the enthalpy of the supply air flowing through the channel. The energy balance of air flowing in the dry channel is given by

$$k_a \frac{d^2 T_{da}^j}{dx^2} - u_{da}^j \rho_{da}^j c_{p,m} \frac{\partial T_{da}^j}{dx} = \frac{h_d^j}{\bar{H}} (T_{da}^j - T_p^j) \quad (1)$$

where, $c_{p,m}$ is the specific heat of air-water vapor mixture and it is defined as

$$c_{p,m} = c_{p,a} + \omega c_{p,v} \quad (2)$$

The superscript j refers to the section number of the channel in the case of four-purge configuration and $\bar{H}$ is the half height of the dry channel. The first term on the left-hand side of Eq.(1) represents the longitudinal heat conduction in the air. The second term denotes the energy variation due to the bulk airflow in the longitudinal direction. The term on the right-hand side of equation indicates the convective heat transfer between the air and the plate surface.

As both sensible and latent cooling effects are involved in the wet channel, heat and mass transfer mechanisms are considered between the air and water film layer, and it is given as follows

$$k_a \frac{d^2 T_{wa}^j}{dx^2} + u_{wa}^j \rho_{wa}^j c_{p,wa} \frac{dT_{wa}^j}{dx} = \frac{h_w^j}{\bar{H}} (T_{wa}^j - T_f^j) + \frac{h_m^j \rho_{da}^j}{\bar{H}} [\omega_{sat}(T_f^j) - \omega_v^j] c_{p,v} (T_{wa}^j - T_f^j) \quad (3)$$

The mass transfer takes place only at the wetted surface, powered by the driving force of vapor partial pressure difference, and its mass exchange is written as

$$\rho_{da} D_v \frac{d^2 \omega_v^j}{dx^2} = -\rho_{da} u_{wa}^j \frac{d\omega_v^j}{dx} + h_m^j \bar{H} \rho_{da} (\omega_{v,sat}(T_f^j) - \omega_v^j), \quad (4)$$

whilst the energy balances for the water film and the impervious layer are given as

$$k_f \frac{d^2 T_f^j}{dx^2} = -\frac{h_w(T_{wa}^j - T_f^j)}{\delta_f} - k_f \frac{T_p^j - T_f^j}{\delta_f^2} + \frac{h_{fg}^j h_m^j \rho_{da} [\omega_{sat}(T_f^j) - \omega_v^j]}{\delta_f} \quad (5)$$

$$k_p \frac{d^2 T_p^j}{dx^2} = -\frac{h_d(T_{da}^j - T_p^j)}{\delta_p} + \frac{k_p(T_p^j - T_f^j)}{\delta_p^2} \quad (6)$$

Given the geometry of the generic cell and the operating conditions, the maximum Reynolds number is 1048, which is sufficiently low that the flow is laminar flow. The convective heat transfer coefficients for the dry and wet channels are determined by Eq. (7).

$$h = \frac{Nu k_a}{D_h} \quad (7)$$

The Nusselt number that at constant surface flux can be calculated by the correlation by Eq.(8) for fully developed laminar flow in the for the rectangular dry channel [40].

$$Nu = 8.235(1 - 2.042AR + 3.085AR^2 - 2.477AR^3 + 1.058AR^4 - 0.186AR^5) \quad (8)$$

where AR is the aspect ratio of the channel (i.e., the ratio of the minimum to the maximum dimensions). The Nusselt number in the wet channel, in which both heat and mass transfer processes are involved, can be calculated by the following equation [41].

$$Nu = 0.10 \left(\frac{l_e}{l} \right)^{0.12} Re^{0.8} Pr^{1/3} \quad (9)$$

where l_e is the characteristic length, $l_e = V/A_s$ and, V is the volume occupied by the wick material or water film. A_s is the wetted surface area of the channel. l is the total thickness of channel plate including the water film.

The convective mass transfer coefficient for the wet channel is approximated using heat and mass transfer analogy and can be expressed by Lewis number.

$$\frac{h}{h_m} = \rho_a c_p Le^{2/3} \quad (10)$$

In addition to the temperature of product air, four different performance indices were adopted to evaluate the performance of the single generic cell of IEC. They are sensible cooling capacity ($\dot{Q}_{sen}$), total cooling capacity of product air ($\dot{Q}_t$), dew-point effectiveness (ε_{dp}) and total efficiency (η_{tot}), which can be calculated by the following formulas

$$\dot{Q}_{sen} = \dot{m}_{sa} c_{p,a} (T_{da,out} - T_{da,in}) \quad (11)$$

$$\dot{Q}_{tot} = \dot{m}_{sa} [i_{in}(T_{da,in}, \omega_{in}) - i_{out}(T_{da,out}, \omega_{in})] \quad (12)$$

where the enthalpy of moist air is determined by both the dry air and the water vapor,

$$i = i_a + \omega i_v = c_{p,a} T_{da} + \omega (c_{p,v} T_{da} + i_o) \quad (13)$$

$$\varepsilon_{dp} = \frac{T_{da,in} - T_{sa}}{T_{da,in} - T_{dp}(T_{da,in})} \quad (14)$$

$$\eta_{tot} = \frac{\dot{Q}_{tot}}{\dot{Q}_{max}} \quad (15)$$

where $\dot{Q}_{max}$ is the maximum cooling capacity and can be expressed as

$$\dot{Q}_{max} = \dot{m}_{da} \left[i_{in} (T_{da,in}, \omega_{in}) - i_{out} (T_{dp}, \omega_{in}) \right] \quad (16)$$

where T_{dp} is the inlet air dew-point temperature of the dry channel.

In this study, we defined the total efficiency other than the dew-point effectiveness in order to analyze the effect of the purge ratio on the total cooling capacity of the product air that includes sensible and latent cooling capacity. The total efficiency is always lower than the dew-point effectiveness as it takes account of the mass flow rates for the primary air, working air and supply air (product air). By using the total efficiency, the maximum available cooling capacity can be estimated with the minimal loss of air mass. $\dot{m}_{sa}$ is the mass flow rate of the supply air, which is determined by the purge ratio. $\dot{m}_{da}$ is the mass flow rate of the air flowing in the dry channel (primary air), which is the sum of the mass flow of working air and the supply air.

Table 1 shows the design and operating parameters for numerical study. The single generic-cell comprises a dry channel, two wet channels, and a separating plate in between them. It should be noted that for the four-purge configuration, the height of the wet channel has different heights in each section in order to maintain the same velocity along the channel.

Table 1. Design and modeling parameters of IEC for numerical study.

Parameter	Values
Channel length (m)	0.6, 0.8, 1.0, 1.2, 1.4
Channel width (m)	0.8
Channel height (mm)	1-purge: 5 4-purge: 5 for dry channel; 7, 5, 2, 4, 1 for wet channel
Purge ratio (%)	1-purge: 20, 25, 30, 35, 40, 45, 50 4-purge: 10, 13, 7, 5
Inlet air velocity (m/s)	2.0
Inlet air dry-bulb temperature (°C)	30
Inlet air humidity ratio (g _{vapor} /kg dry air)	10

The mesh grid size is set at 1.0 mm when solving the differential equations. For instance, 1.0 m channel has 1000 cell elements. The governing equations (from Eq. (1) to Eq. (6)) are solved using the solver (bvp4c) for coupled-nonlinear ordinary differential equations in MATLAB environment.

4. Results and discussions

4.1 Model validation

As shown in Fig. 1 (a), each dry channel is placed in between two wet channels and each wet channel is adjacent to two dry channels. In this study, our differential control volume contains half the height of the dry and wet channels by considering each channel is symmetry about the horizontal centerline since the air flows are assumed to be fully developed. Our model has been validated with the published experimental results from references [42] and [43] where a series of experiments were carried out on a dew-point evaporative cooler with

two pairs of dry and wet channels. To evaluate the accuracy of mathematical models, root-mean-square error (RMSE) was used for the consistency between simulated results and experimental data, and it is expressed as [44]

$$RMSE = \sqrt{\frac{\sum [(X_{sim,i} - X_{exp,i}) / X_{exp,i}]^2}{n}} \quad (17)$$

Fig. 2 (a) shows the comparison of the temperature distribution along the channel between the simulation results and the experimental results under the specific inlet air conditions of 32.6 °C temperature with 14.0 g_{vapor}/kg_{dry air} humidity ratio and 38.2 °C temperature with 10.2 g_{vapor}/kg_{dry air} humidity ratio. It is observed that the maximum deviation of the model is 1.9 % with RMSE of 1.1 % under inlet temperature of 32.6 °C, and 3.2 % with RMSE of 1.6 % under inlet temperature 38.2°C. Fig. 2 (b) also shows a good agreement between the simulation and experimental data under various inlet temperatures and humidity ratios. The maximum deviation of 3.3 % is observed at the inlet air temperature of 26°C with 26.0 g_{vapor}/kg_{dry air} humidity ratio, and the RMSE value is 2.0%.

Fig. 2 Model validation with experimental data from references [42,45]: (a) Temperature distributions along the channel under specific primary air conditions ($T_{in} = 32.6$ °C with $\omega = 14.0$ g_{vapor}/kg_{dry air}; and $T_{in} = 38.2$ °C with $\omega = 10.2$ g_{vapor}/kg_{dry air}.) (b) Product air temperature under various inlet air temperatures (25, 30, 35, 40 and 45°C) and humidity ratios (6.9, 11.2, 20 and 26 g_{vapor}/kg_{dry air}).

4.2 Comparative analysis on the performance of a single-purge and four purge configurations

Based on the mathematical models, comparative analyses were carried out to identify the differences between the single-purge and four-purge configurations. The four-purge configuration is divided into two different types. In type A, the working air is purged from the end of each section of the dry channel and then added to the preceding working air stream with assorted purge ratios (15%, 13%, 7%, and 5% of the primary air) as shown in Fig.1 (b). For type B, the working air is purged from the end of the channel into another small passage, from which the purged air is distributed into each section of the wet channel with the same purge ratios with type A. It is worthy to note that all the distributed purged air has the same temperature as the outlet air temperature of the dry channel. In this study, it is assumed that the airflow rates are controlled by using differently sized diaphragms and custom cut orifices plate.

Fig. 3 Comparisons of temperature and humidity ratio profiles between one- purge and four purge configurations: (a) temperature profiles of type A, (b) temperature profiles of type B, (c) humidity profiles of type A, (d) humidity profiles of type B.

Fig. 3 depicts the temperature and humidity ratio profiles of the product air and the working air for the four-purge IEC units with two different schemes. Those of the single-purge configuration are also plotted as a benchmark. It is observed from Figs. 3 (a) and 3 (b) that the single-purge configuration produces the product air of lower temperature than the four-purge configurations. The blue dotted line is the product air within the each section and the red solid line indicates the corresponding working air along the each section. It is also observed that for type A, the inlet temperature of the working air slightly drops at the entrance of each section while it increases in type B. It can be seen from Figs 3 (c) and 3 (d) that the saturation of working air appears only at the exits of last two sections. The blue line is the humidity ratio of air in the wet channel and the red line indicate the corresponding saturated humidity ratio of the air within the section. From the results, it is apparent that a longer contact time is essential for greater mass transfer to take place and thus greater heat transfer for both configurations.

Fig. 4 shows the dry-bulb temperature and humidity profiles of four-purge configurations under the baseline condition where the inlet air temperature is 30°C and humidity ratio is 10 $g_{\text{vapor}}/kg_{\text{dry air}}$. The larger decrease in temperature is observed in the nearer section to the inlet of the dry channel. This is mainly attributed to the flow rate of the working air and the difference of inlet temperature between two streams. It is also observed that the working air of the first section is not saturated compared to those of other sections.

Fig. 4 Temperature and humidity profiles on psychrometric chart for four-purge configurations: (a) type A, (b) type B

Fig. 5 Performance comparisons for a generic cell with different purge configurations: (a) outlet air temperature, (b) sensible and total cooling capacities, dew-point effectiveness, and total efficiency.

Fig. 5 shows outlet air temperature, cooling capacities (sensible cooling capacity and total cooling capacity) and total efficiency of a generic cell with different purge configurations. The baseline weather condition was applied to the simulation process. The total working-to-intake air ratio (purge ratio) is 0.4 for all the cases. It can be seen that the outlet air temperature of the single-purge configuration is 20.7°C, which is 2.1°C lower than that of the type B. For the single-purge configuration, dew-point effectiveness reaches 0.58, which is 20% higher than that of type B and 30% higher than type A. Meanwhile, total efficiency of the single-purge configuration is 0.38, which is 20% higher than that of type B and 30% higher than type A. Total cooling capacity of the single-purge configuration is 59.15W while the four-purge configurations achieve 49.12W and 45.5W respectively. It should be noted that the total cooling capacity is slightly higher than the sensible cooling capacity since total cooling capacity accounts the cooling effect of water vapor other than the sensible cooling capacity.

4.3 Effects of channel length and purge ratio for single-purge configuration

Figs. 6 (a) and (b) depict the dry bulb temperature and humidity profiles on psychrometric chart for different channel lengths and different purge ratios, respectively. The inlet air temperature and humidity ratio were kept at 3 °C and 10 g_{vapour}/kg_{dry air}, respectively. The channel length varied from 0.6 to 1.4 m while keeping the purge ratio of 40%. When the purge ratio increased from 20 to 50%, the channel length was fixed at 1.0 m. It is observed that the temperature in the dry channel gradually decreases as the channel length and the purge ratio increase. The product air temperature reaches a minimum of 19.0°C when the channel length is 1.4 m and the purge ratio is 50%. It can be easily seen that the working air is cooled down and saturated towards its saturation line (100%) and then the temperature increases while evaporation takes place continuously.

Fig. 6 Temperature and humidity profiles of single-purge configuration under baseline condition: (a) different channel lengths of 0.6, 0.8, 1.0, 1.2 and 1.4m, (b) different purge ratios of 20, 25, 30, 35, 40, 45 and 50%, (c) humidity ratio and saturated humidity ratio profiles of different channel lengths.

Fig.6 (c) shows that the humidity ratio of the working air steadily increases along the flow direction even after it has almost reached its saturated level. This is because the saturated humidity ratio also grows continuously at the same rate along the flow direction with the increasing the temperature. As a result, continuous evaporation ensues along the wet channel. It is also observed that the saturation point where the working air reaches its saturation level is the same and independent on the channel length. Therefore, it is apparent that a contact

time of the working air affects the amount of evaporation, which in turn leads to larger temperature drop.

Fig. 7 (a) shows the effect of the channel length on the product temperature of the single-purge configuration when the purge ratio was kept at 40%. It is found that the outlet temperature is largely affected by the channel length. The outlet temperature of the product air drops from 21.3 to 19.01°C as the channel length increases from 0.6 to 1.4 m. Fig. 7 (b) shows both sensible and total cooling capacities and dew-point effectiveness and total efficiency of the single-purge configuration with different channel lengths under baseline condition. It can be seen that both cooling capacities, dew-point effectiveness and total efficiency increase as the channel length increases due to a longer contact time of the working air stream.

Fig. 7 Effect of channel length on the performance of single-purge configuration under baseline condition: (a) product air temperature, (b) sensible and total cooling capacities, dew-point effectiveness, and total efficiency.

Fig. 8 (a) shows the effect of the purge ratio on the outlet temperature of the product air. It is found that the product air temperature is also strongly affected by the purge ratio. The temperature drops from 23.8 to 19.01°C as the purge ratio increases from 20 to 50%. Fig. 8 (b) shows cooling capacity and efficiency of a single-generic cell with different purge ratios. It is observed that the sensible and total cooling capacities increase until the purge ratio reaches 35%, and thereafter it gradually drops. The same trend is also observed for the total efficiency. The maximum efficiency appears at 35% purge ratio. However, the dew-point effectiveness increases as the purge ratio increase. This is because the dew-point effectiveness is only related to the temperature difference whereas the total efficiency is concerned with the mass flow rate of the product air as well as the temperature difference.

Fig. 8 Effect of purge ratio on the performance of single-purge configuration under baseline condition: (a) outlet air temperature, (b) sensible and total cooling capacities, dew-point effectiveness, and total efficiency.

Fig. 9 shows the product air temperature, the total cooling capacity and the dew-point effectiveness of single-purge configuration when the channel length varies from 0.6 to 1.4 m, and the purge ratio ranges from 20 to 50%. It can be easily seen that lower product air temperature, as well as higher dew-point effectiveness can be achieved with longer channel length and larger purge ratio. However, the maximum cooling capacity can be achieved with longer channel length and 35% purge ratio.

Fig. 9 Influence of channel length and purge ratio on the performance of single-purge configuration: (a) product air temperature, (b) dew-point effectiveness, (c) total cooling capacity.

5. Conclusion

A detailed mathematical model has been established to study a counter-flow indirect evaporative cooler with two different purge ratios. Validation of simulation results with experimental data obtained from references demonstrated that the model yields satisfactory outcome with remarkable accuracy. The key findings from this study include:

- (1) For type A of the four-purge configuration, the inlet temperature of the working air slightly drops at the entrance of each section while it increases in type B. For the four-purge configuration, the saturation of working air appears only at the exits of last two sections. Therefore, a longer contact time is essential for greater mass transfer to take place and thus greater heat transfer.
- (2) The larger decrease in temperature is observed in the nearer section to the inlet of the dry channel of the four-purge configuration. This is mainly attributed to the flow rate of the working air and the difference of inlet temperature between two streams.
- (3) The single-purge configuration produces a lower product air temperature, around 10% lower than two four-purge configurations. It can also achieve 58% of dew-point effectiveness, 20% higher than type B and 30% higher than type A. Total cooling capacity of the single-purge configuration is 59.15 W while the four-purge configurations achieve 49.12W and 45.5W respectively.
- (4) For the parameter study on the single-purge configurations, lower product air temperature and higher dew-point effectiveness were achieved with longer channel length and larger purge ratio. However, the maximum cooling capacity approached its maximum with longer channel length and 35% purge ratio. This is because the dew-point effectiveness is only related to the temperature difference whereas the total efficiency is concerned with the mass flow rate of the product air as well as the temperature difference.

Based on the detailed simulations, it is noted that the IEC have great potential for air conditioning of commercial and residential buildings. It would lead to significant savings in

both electricity consumption as well as water consumption, vis-à-vis to the conventional vapor compression chillers with water cooling towers. It can be estimated that the kW/RT of IEC can be reduced to less than 0.5 ± 0.03 versus the existing average kW/RT of about 0.85 ± 0.03 . In addition, IEC approach has low capital cost due primarily to the avoided hardware needed.

Nomenclature

<i>AC</i>	air conditioning
<i>CGAR</i>	compound annual growth rate
c_p	specific heat at constant pressure, J/(kg·K)
D_v	diffusion coefficient, m ² /s
<i>EER</i>	energy efficiency ratio
<i>GHG</i>	greenhouse gas
h	convective heat transfer coefficient, W/(m ² ·K)
h_{fg}	latent heat evaporation, J/kg
h_m	mass transfer coefficient, m/s
$\bar{H}$	half height of channel, m
<i>HVAC</i>	heating, ventilation, and air conditioning
i	enthalpy, J/kg
<i>IEC</i>	indirect evaporative cooler
k	thermal conductivity, W/(m·K)
L	channel length, m
$\dot{m}$	mass flow rate, kg/s
P	pressure, Pa
$\dot{Q}$	heat transfer rate, W
<i>RMSE</i>	Root-mean-square error
r	working air ratio
T	temperature, °C
u	velocity, m/s
W	channel width, m
x	x-coordinate, m

Greek symbols

δ	thickness, mm
ρ	density, kg/m ³
ω	humidity ratio, kg/kg dry air

Subscripts

a	air
da	dry channel air
dp	dew point
f	water film
in	inlet
m	water vapor-air mixture
out	outlet
p	plate
pa	primary air
sa	supply air
sat	saturated

v water vapor
wa wet channel air

Acknowledgement

This research was supported by King Abdullah University of Science and Technology (KAUST) and Korean National Research Foundation (NRF-2017R1A2A1A05001461)

Reference

- [1] Future Market Insights. Refrigeration and Air Conditioning Compressors Market: Global Industry Analysis 2013 – 2017 and Opportunity Assessment, 2018 – 2028. 2018.
- [2] Shahzad MW, Burhan M, Ang L, Ng KC. Energy-water-environment nexus underpinning future desalination sustainability. *Desalination* 2017;413:52–64. doi:10.1016/j.desal.2017.03.009.
- [3] Shahzad MW, Burhan M, Ghaffour N, Ng KC. A multi evaporator desalination system operated with thermocline energy for future sustainability. *Desalination* 2018;435:268–77. doi:10.1016/j.desal.2017.04.013.
- [4] International Energy Agency (IEA). the Future of Cooling 2018:92.
- [5] La D, Li Y, Dai YJ, Ge TS, Wang RZ. Development of a novel rotary desiccant cooling cycle with isothermal dehumidification and regenerative evaporative cooling using thermodynamic analysis method. *Energy* 2012;44:778–91. doi:10.1016/j.energy.2012.05.016.
- [6] Waite M, Cohen E, Torbey H, Piccirilli M, Tian Y, Modi V. Global trends in urban electricity demands for cooling and heating. *Energy* 2017;127:786–802. doi:10.1016/j.energy.2017.03.095.
- [7] Izquierdo M, Moreno-Rodríguez A, González-Gil A, García-Hernando N. Air conditioning in the region of Madrid, Spain: An approach to electricity consumption, economics and CO₂ emissions. *Energy* 2011;36:1630–9. doi:10.1016/j.energy.2010.12.068.
- [8] Electricity & Cogeneration Regulatory Authority (ECRA). Annual Report 2014:145.
- [9] Oh SJ, Ng KC, Thu K, Chun W, Chua KJE. Forecasting long-term electricity demand for cooling of Singapore's buildings incorporating an innovative air-conditioning technology. *Energy Build* 2016;127:183–93. doi:10.1016/j.enbuild.2016.05.073.
- [10] SJ Oh, K Choon Ng, K Thu, M Kum Ja, MR Islam, W Chun KC. Studying the performance of a dehumidifier with adsorbent coated heat exchangers for tropical climate operations.pdf. *Sci Technol Built Environ* 2017;23:127–35.
- [11] Oh SJ, Ng KC, Chun W, Chua KJE. Evaluation of a dehumidifier with adsorbent coated heat exchangers for tropical climate operations. *Energy* 2017;137:441–8. doi:10.1016/j.energy.2017.02.169.
- [12] Todesco G. Chillers+ Lighting+ TES. *Ashrae J* 2005;47:18–25.
- [13] Chua KJE. Low Energy Air Conditioning. Summ. Rep. Conf. Energy Sustain. Cities, Singapore: 2012.
- [14] Hong A, Chapter K, Workshop T. Fundamentals of Water System Design 17, 18, 24, 25. Development 2007.
- [15] Duan Z, Zhan C, Zhang X, Mustafa M, Zhao X. Indirect evaporative cooling : Past , present and future potentials. *Renew Sustain Energy Rev* 2012;16:6823–50. doi:10.1016/j.rser.2012.07.007.
- [16] Mahmood MH, Sultan M, Miyazaki T, Koyama S, Maisotsenko VS. Overview of the

- Maisotsenko cycle – A way towards dew point evaporative cooling. *Renew Sustain Energy Rev* 2016;66:537–55. doi:10.1016/j.rser.2016.08.022.
- [17] Lin J, Thu K, Bui TD, Wang RZ, Ng KC, Chua KJ. Study on dew point evaporative cooling system with counter-flow configuration. *Energy Convers Manag* 2016;109:153–65. doi:10.1016/j.enconman.2015.11.059.
- [18] Lee J, Lee D. International Journal of Heat and Mass Transfer Experimental study of a counter flow regenerative evaporative cooler with finned channels. *Int J Heat Mass Transf* 2013;65:173–9. doi:10.1016/j.ijheatmasstransfer.2013.05.069.
- [19] Gao WZ, Cheng YP, Jiang AG, Liu T, Anderson K. Experimental investigation on integrated liquid desiccant e Indirect evaporative air cooling system utilizing the Maisotesenko e Cycle. *Appl Therm Eng* 2015;88:288–96. doi:10.1016/j.applthermaleng.2014.08.066.
- [20] Comino F, Ruiz de Adana M, Peci F. Energy saving potential of a hybrid HVAC system with a desiccant wheel activated at low temperatures and an indirect evaporative cooler in handling air in buildings with high latent loads. *Appl Therm Eng* 2018;131:412–27. doi:10.1016/j.applthermaleng.2017.12.004.
- [21] Chen Y, Yang H, Luo Y. Investigation on solar assisted liquid desiccant dehumidifier and evaporative cooling system for fresh air treatment. *Energy* 2018;143:114–27. doi:10.1016/j.energy.2017.10.124.
- [22] Sohani A, Sayyaadi H. Thermal comfort based resources consumption and economic analysis of a two-stage direct-indirect evaporative cooler with diverse water to electricity tariff conditions. *Energy Convers Manag* 2018;172:248–64. doi:10.1016/j.enconman.2018.07.008.
- [23] Sohani A, Farasati Y, Sayyaadi H. A systematic approach to find the best road map for enhancement of a power plant with dew point inlet air pre-cooling of the air compressor. *Energy Convers Manag* 2017;150:463–84. doi:10.1016/j.enconman.2017.08.028.
- [24] Du Y, Gai W mei, Jin L zhe, Sheng W. Thermal comfort model analysis and optimization performance evaluation of a multifunctional ice storage air conditioning system in a confined mine refuge chamber. *Energy* 2017;141:964–74. doi:10.1016/j.energy.2017.09.123.
- [25] Xu P, Ma X, Zhao X, Fancey K. Experimental investigation of a super performance dew point air cooler. *Appl Energy* 2017;203:761–77. doi:10.1016/j.apenergy.2017.06.095.
- [26] Duan Z, Zhan C, Zhao X, Dong X. Experimental study of a counter- fl ow regenerative evaporative cooler. *Build Environ* 2016;104:47–58. doi:10.1016/j.buildenv.2016.04.029.
- [27] Kim HJ, Ham SW, Yoon DS, Jeong JW. Cooling performance measurement of two cross-flow indirect evaporative coolers in general and regenerative operation modes. *Appl Energy* 2017;195:268–77. doi:10.1016/j.apenergy.2017.03.053.
- [28] Hasan A. Going below the wet-bulb temperature by indirect evaporative cooling : Analysis using a modified e -NTU method. *Appl Energy* 2012;89:237–45. doi:10.1016/j.apenergy.2011.07.005.
- [29] Sadighi Dizaji H, Hu EJ, Chen L, Pourhedayat S. Development and validation of an analytical model for perforated (multi-stage) regenerative M-cycle air cooler. *Appl Energy* 2018;228:2176–94. doi:10.1016/j.apenergy.2018.07.018.
- [30] Lin J, Bui DT, Wang R, Chua KJ. The counter-flow dew point evaporative cooler: Analyzing its transient and steady-state behavior. *Appl Therm Eng* 2018;143:34–47. doi:10.1016/j.applthermaleng.2018.07.092.

- [31] Jafarian H, Sayyaadi H, Torabi F. Modèle numérique d'un refroidisseur évaporatif indirect à contre-courant à point de rosée avec une condition aux limites modifiées et prenant en considération les effets des zones d'entrée. *Int J Refrig* 2017;84:36–51. doi:10.1016/j.ijrefrig.2017.09.003.
- [32] Comino F, Milani S, De Antonellis S, Joppolo CM, Ruiz de Adana M. Simplified performance correlation of an indirect evaporative cooling system: Development and validation. *Int J Refrig* 2018;88:307–17. doi:10.1016/j.ijrefrig.2018.02.002.
- [33] Hasani Balyani H, Sohani A, Sayyaadi H, Karami R. Acquiring the best cooling strategy based on thermal comfort and 3E analyses for small scale residential buildings at diverse climatic conditions. *Int J Refrig* 2015;57:112–37. doi:10.1016/j.ijrefrig.2015.04.008.
- [34] Zhao X, Yang S, Duan Z, Riffat SB. Feasibility study of a novel dew point air conditioning system for China building application. *Build Environ* 2009;44:1990–9. doi:10.1016/j.buildenv.2009.02.003.
- [35] Duan Z, Zhao X, Zhan C, Dong X, Chen H. Energy saving potential of a counter-flow regenerative evaporative cooler for various climates of China : Experiment-based evaluation. *Energy Build* 2017;148:199–210. doi:10.1016/j.enbuild.2017.04.012.
- [36] Sohani A, Sayyaadi H, Mohammadhosseini N. Comparative study of the conventional types of heat and mass exchangers to achieve the best design of dew point evaporative coolers at diverse climatic conditions. *Energy Convers Manag* 2018;158:327–45. doi:10.1016/j.enconman.2017.12.042.
- [37] Sohani A, Sayyaadi H, Hoseinpoori S. Modeling and multi-objective optimization of an M-cycle cross-flow indirect evaporative cooler using the GMDH type neural network. *Int J Refrig* 2016;69:186–204. doi:10.1016/j.ijrefrig.2016.05.011.
- [38] Bi Y, Wang Y, Ma X, Zhao X. Investigation on the energy saving potential of using a novel dew point cooling system in data centres. *Energies* 2017;10:1–21. doi:10.3390/en10111732.
- [39] Metzger I. Multistaged Indirect Evaporative Cooler Evaluation 2014.
- [40] Shah, R.K. and London AL. *Laminar Flow Forced Convection in Ducts*. Acad Press New York 1978.
- [41] J.A. Dowdy, P.E. PD. NSK. Experimental determination of heat and mass transfer coefficients in rigid impregnated cellulose evaporative media. *Ashrae Trans* 1987.
- [42] Lin J, Bui DT, Wang R, Chua KJ. On the fundamental heat and mass transfer analysis of the counter-flow dew point evaporative cooler. *Appl Energy* 2018;217:126–42. doi:10.1016/j.apenergy.2018.02.120.
- [43] Knudsen M. Die maximale Verdampfungsgeschwindigkeit des Quecksilbers. *Ann Phys* 1915;352:697–708. doi:10.1002/andp.19153521306.
- [44] Chai T, Draxler RR, Prediction C. Root mean square error (RMSE) or mean absolute error (MAE)? – Arguments against avoiding RMSE in the literature 2014:1247–50. doi:10.5194/gmd-7-1247-2014.
- [45] Rianguilaikul B, Kumar S. An experimental study of a novel dew point evaporative cooling system. *Energy Build* 2010;42:637–44. doi:10.1016/j.enbuild.2009.10.034.