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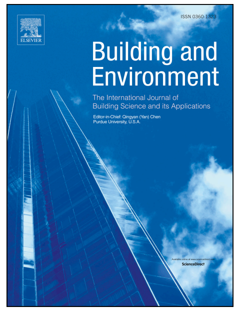

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Experimental study of a counter-flow regenerative evaporative cooler

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Abstract:

This paper aims to investigate the operational performance and impact factors of a counter-flow regenerative evaporative cooler (REC). This was undertaken through a dedicated experimental process. Temperature, humidity and flow rate of the air flows at the inlet, outlet and exhaust opening of the cooler were tested under various operational conditions, i.e., different inlet air conditions, feed water temperature and evaporation rate were also correspondingly measured. It was found that the wet-bulb effectiveness of the presented cooler ranged from 0.55 to 1.06 with Energy Efficiency Ratio (EER) rated from 2.8 to 15.5. The major experimental results were summarised below: 1) the wet-bulb effectiveness was significantly enhanced through either ways of increasing inlet wet-bulb depression or reducing intake air velocity, or alternatively

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by increasing working-to-intake air ratio; 2) the cooling capacity and EER of cooler was rapidly increased by means of increasing inlet wet-bulb depression or increasing intake air velocity, or reducing working-to-intake air ratio instead; 3) the effectiveness reduced by less 5% while feed water temperature increased from 18.9 to 23.1 ^{o}C; 4) apparent acceleration in water evaporation rate was gained from increasing inlet wet-bulb depression or air velocity. The presented cooler showed 31% increase in wet-bulb effectiveness and 40% growth in EER compared to conventional indirect evaporative cooler. The research helped identifying the performance of a new REC with enhanced performance and thus contributed to development of energy efficient air conditioning technologies, which eventually lead to significant energy saving and carbon emission reduction in air conditioning sector.

Keywords: Air-conditioning; Evaporative cooling; Heat and mass transfer; Experimental study

Nomenclature

c_p specific heat at constant pressure, kJ/kg K

h enthalpy of moist air, kJ/kg

m mass flow rate, kg/s

P power consumption, W

Q cooling capacity, W

t temperature, °C

u air velocity in channels, m/s

V airflow rate, m^3/h

V_W water evaporation rate, kg/h or litre/h

w humidity ratio, kg/kg dry air

Greek letters

ε cooling effectiveness, %

ρ density, kg/m^3

Subscript

1 intake air of dry channels

2 product air of dry channels

3 exhaust air of wet channels

a dry air

db dry-bulb

dp dew-point

w water film

wb wet-bulb

Abbreviations

DB dry bulb

DP dew point

EER Energy Efficiency Ratio

exp experiment

sim simulation

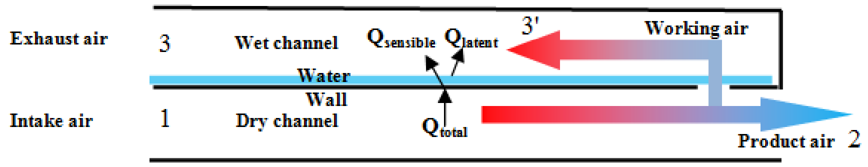
temp temperature

WB wet bulb

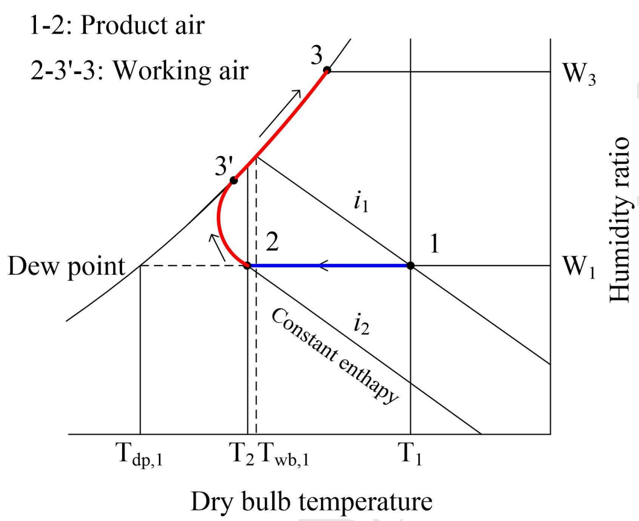
1. Introduction

Most of air-conditioning systems currently in applications are based on mechanical vapour compression refrigeration cycle for the advantages of good stability in performance, low initial cost, and long life cycle time. However, the mechanical vapour compression system not only consumes large quantities of electricity due to continuous operation of compressor, but also affects environment negatively by ways of contributing to the greenhouse effect as producing carbon emissions as well as damaging the ozone layer owing to the use of CFCs and HCFCs refrigerants [1,2]. Without compressor and refrigerant participating, evaporative cooling is considered to be a promising alternative through entirely replacing mechanical compression system [3,4] or adding as a pre-cooler for the conventional mechanical system [5–7]. This cooling method utilises the principle of water evaporation for absorbing heat existing in ambient, therefore, consuming much lower electricity compared to mechanical compression system.

Direct Evaporative Cooler (DEC) and Indirect Evaporative Cooler (IEC) are the two types of evaporative coolers commonly used nowadays. For DEC, product air keeps contact with water directly, causing evaporation of the water, giving rise to reduction of temperature and moisture rise of the product air simultaneously. For indirect evaporative cooler, the temperature of product air can be lowered without adding humidity [8] that makes the indirect cooler more attractive than the direct one. However, the wet-bulb effectiveness of conventional IEC is only in the range of 55-75% [9]. Because the direct air-water contact process taking place at the wet side of conventional IEC is limited by the ultimate wet bulb temperature of the entering air. Therefore, the conventional IEC cannot cool the entering air below its wet-bulb temperature leading to low cooling capacity, which cannot satisfy increasing demand of building air conditioning. Regenerative evaporative cooler (REC) is a type of indirect evaporative cooling system which has the potential of lowering the intake air temperature below the wet-bulb close to its dew point temperature without any moisture increased [10]. The airflow path of REC is shown in Fig. 1(a) schematically. In the configuration, portion of intake air in dry channel is redirected into adjacent wet channel through the small perforations at the end of dry channel. The thermal process of airflows in dry and wet channels of REC is represented on psychrometric chart as shown in Fig. 1(b). The airstream in dry channel experiences constant moisture cooling process (From state 1 to 2) and is cooled towards dew-point temperature of intake air (T_{dp}) due to enhanced heat and mass transfer between the product and working air. The maximum heat transfer is caused because the inlet temperature of working air (fraction of intake air) has been fully pre-cooled towards dew point of intake air. In the wet channel, meanwhile, because of water evaporation, the working air experiences a constant enthalpy cooling process to become saturated (From state 2 to 3'). Continuously, saturated working air exchanges heat with the air in dry channel sensibly and latently until being discharged (From state 3' to 3). In this process, both the latent and sensible heat transfer has been occurred simultaneously.



(a)



(b)

Fig. 1 Working principle of regenerative evaporative cooler (a) Schematic of airflow paths (b) thermal process of air treatment representing on psychrometric chart

Over the last few decades, numerous numerical studies have been performed for predicting the thermal performance of REC system under various climatic conditions and analysing the effects of operating and geometrical parameters. Maclaine-cross and Banks [11] developed a linear approximate model of wet surface heat exchangers and used it to predict the performance of a proposed regenerative evaporative cooler using such an exchange. Hsu and Lavan [12] numerically investigated several types of wet-surface heat exchanger including unidirectional, counter-flow, counter-flow regenerative, cross-flow regenerative configurations. They found that inlet dew point temperature can be approached for cross-flow and counter-flow regenerative configurations. Maisotsenko proposed a new thermodynamic cycle (M-cycle) and developed a cross-flow heat exchanger configuration for regenerative evaporative cooler based on the M-cycle [13]. Zhan et al. [14,15] carried out a numerical parametric study of the cooling performance of the counter-flow and cross-flow REC heat exchangers based on M-cycle. The numerical model predicted that the counter-flow REC offered greater (around 20% higher) cooling capacity, as well as greater (15%-23% higher) effectiveness when equal in physical size and under the same operating conditions with cross-flow REC. The cross-flow system, however, had a greater (10% higher) EER. Cui et al. [16] numerically analysed the impacts of operating conditions and geometric parameters on performance of a REC heat exchanger based on counter-flow configuration. They also developed a modified log mean temperature difference (LMTD) method for predicting thermal performance of the presented counter-flow REC and M-cycle cross-flow REC [17]. Hasan [18] proposed a modified ε-NTU analytical method to achieve sub-wet bulb temperature by indirect evaporative cooling of air. They applied the analytical method to determine the performance of a counter-flow regenerative evaporative cooler. Pandelidis et al. [19] compared the cross-flow and counter-flow regenerative evaporative cooling heat and mass exchanger by developing a modified ε-NTU model and analysed the influence of geometric and operational parameters on the cooling performance of units. Their developed model was verified against the data obtained both from the modified ε-NTU model created by Hasan [18] and the numerical model developed by Zhan [14]. Anisimov et al. [20] conducted a two-dimensional ε-NTU modelling study for their presented cooler combining parallel and regenerative counter-flow arrangement. The analytical model was validated against the experimental data acquired from Riangvilaikul and Kumar [21]. Their study compared the performance of present cooler with that of conventional regenerative unit and the result showed that the presented unit was able to achieve higher effectiveness. Heidarinejad and Moshari [22] presented a numerical model of a cross-flow indirect evaporative cooler with consideration of wall longitudinal heat conduction (LHC) and effects of spray water temperature variation. They used the model to compare the effectiveness of both two-stage IEC/DEC system and counter-flow REC system against one-stage IEC system. The results showed that both systems have more than 50% higher wet-bulb effectiveness.

A few experimental studies have been conducted for validating numerical models and acquiring the operating performance of REC system under various intake conditions. The performance of a M-cycle indirect evaporative cooling unit manufactured by Coolerado Company was measured [23]. The wet-bulb effectiveness of the unit ranged from 81% to 91% for all test conditions. Riangvilaikul and Kumar [21] presented an experimental study of a counter-flow regenerative evaporative cooler in rectangular configuration operating under various inlet air conditions covering dry, temperate and humid climates. The dew point effectiveness of their developed REC ranged from 0.58 to 0.84 for various inlet conditions, higher than cross-flow type REC. Bruno [24] carried out an on-site experimental testing into the operational characteristics of a counter-flow flat-plate REC unit installed both for commercial and residential applications under various ambient conditions. As a pre-cooler of a refrigeration air conditioner in the commercial application, the REC unit could achieve a wet-bulb effectiveness of 93 to 106% while for residential application, the wet-bulb effectiveness was ranged from 118 to 129%. Lee et al. [25] investigated a method to improve the compactness of the REC by comparing three different configurations, i.e. the flat plate type, corrugated plate type and the finned channel type. Among the considered types, the finned channel type was found to have the smallest volume. An experimental study was conducted to investigate the thermal performance of a counter flow regenerative evaporative cooler with finned channel. The experimental results showed that the wet-bulb effectiveness of the cooler was about 1.2 when intake air temperature and relative humidity is 32 ^{o}C and 50% [26]. Zube and Gillan [27] conducted an experimental study of the cross-flow M-cycle heat exchanger to obtain the internal parameters for product and exhaust channels. Jradi and Riffat [28] conducted an experimental and numerical investigation of an indirect evaporative cooling system being able to achieve dew point cooling. They developed a two-dimensional numerical model for predicting the M-cycle cross-flow REC heat and mass exchanger and analysing effects of operational parameters.

Most recently, a few studies have been conducted to investigate the performance of hybrid systems integrating indirect evaporative coolers with desiccant systems or mechanical vapour compression systems. By using indirect evaporative cooler, energy consumption of the combined systems can be reduced dramatically. To investigate the performance an IEC as a pre-cooling unit for hybrid system in hot and humid climate, Chen et al. [29] developed a numerical model of IEC taking the condensation from primary air into consideration and conducted parameter analysis under three different condensation conditions. The results showed that the condensation lowers the cooling efficiency of IEC but improves the heat transfer rate due to dehumidification. Gao et al. [30] conducted a numerical study of a solid desiccant cooling system with indirect evaporative cooler. The effects of the operating parameters and NTU on the performance of both heat exchanger and IEC were numerically investigated. Saghafifar and Gadalla [31] analysed the performance of two integrated solar PV/T desiccant air conditioning systems using M-cycle regenerative cooler under different operating conditions. Kim et al. [32] developed a practical model for predicting the thermal performance of a wet-coil IEC through statistical analysis of cooling and heat exchange effectiveness data generated by ε-NTU method. The energy saving potential by integrating a wet-coil IEC into variable air volume (VAV) for pre-conditioning incoming outdoor air was also evaluated. Cui et al. [7] developed a computational model to numerically investigate the performance of a hybrid evaporative pre-cooling system with condensation from the product air by employing the room exhaust air as the working air under humid conditions. The simulation results indicated that the proposed hybrid system can remove 47% of the cooling load for humid climate with only a small amount of fan power consumption.

An extensive review has been conducted to explore the recent status and developments of indirect evaporative cooling technologies. It has shown that there is no reported testing work has been taken to investigate the effects of various operating parameters on the overall operational performance of counter-flow REC. Therefore, in this paper, a prototype of counter-flow regenerative evaporative cooler has been developed based on our numerical results [33]. Various sets of dedicated experimental testing were undertaken to investigate the operational performance and impact factors of the cooler, including inlet air conditions (temperature, humidity and velocity), working-to-intake air flow ratio and feed water temperature.

2. Description and performance indicators of regenerative evaporative cooler

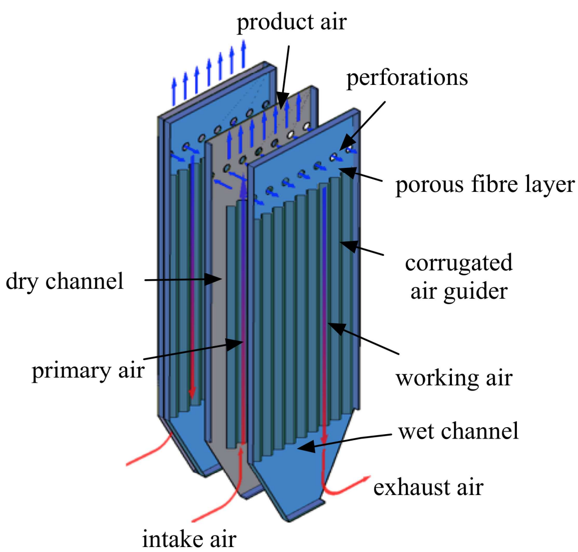
2.1 Description and fabrication of regenerative evaporative cooler

In this study, a counter-flow heat and mass exchanger (HMX), as the core module of regenerative evaporative cooler, has been designed and fabricated in laboratory. The HMX module was stacked together using multiple groups of dry and wet channels, as demonstrated in Fig. 2(a). The dry and wet channels were separated by a thin aluminium flat plate and both channels were supported by plastic corrugated sheets as the guiders for product and working air. The geometry of corrugated sheets is schematically shown in Fig. 2(b). Porous fibre layer was applied to the internal surface of wet channels to improve surface wettability. The whole HMX module is divided into 5 small groups. Each group includes 9 dry channels and 10 wet channels. The geometric parameters of dry channels, wet channels and the corrugated sheets are specified in Table 1.

Table 1 Geometric size for dry channels, wet channels and corrugated sheets of fabricated HMX module

Parameters	Unit	Size
Channel length	m	0.9
Channel width	mm	314
Plate thickness	mm	0.25
Channel spacing	mm	6
Length of corrugated sheets	m	0.9
Width of corrugated sheets	mm	314
Thickness of Corrugated sheets	mm	0.2
Height of corrugated sheets	mm	5.8
Pitch of corrugated sheets	mm	12

A prototype of regenerative evaporative cooler consisting of the HMX module has been developed. The demonstration of the cooler is schematically shown in Fig. 2(c). The cooler operates in the following way: The intake air of the cooler would be dragged into dry channels from the bottom left side of the HMX module by supply air fan. The air flows through dry channels and is divided into two parts at the end of the channels: One part of the airstream, i.e. product air keeps moving at the same direction and is finally delivered to the space where cooling is required, and the other part of the airstream, i.e. working air, is extracted by another fan to the adjacent wet channels where the inner surfaces are wetted by water. The wet channel allows heat to be absorbed through the channel wall by vaporising water on the surface. The working air in the wet channel flows in a reverse direction and is finally discharged to ambient from the bottom right side of the cooler.



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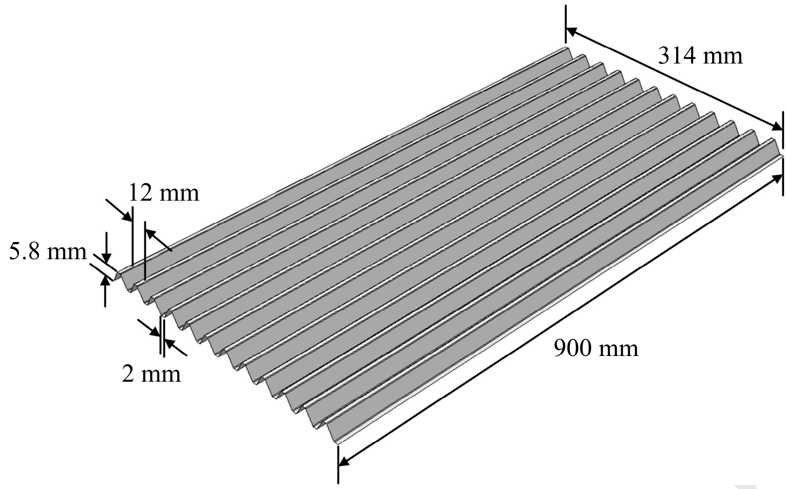

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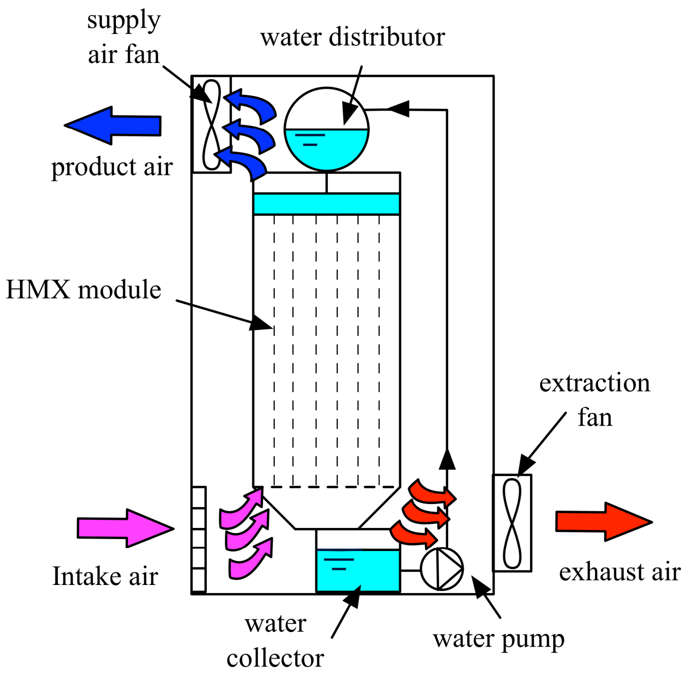


(c)

Fig. 2 Schematics of proposed regenerative evaporative cooler (a) configuration of dry and wet channels (b) geometry of corrugated sheets (c) prototype of regenerative evaporative cooler

2.2 Performance indicators

2.2.1 Cooling effectiveness

The wet-bulb effectiveness of REC is the ratio of the difference between intake and outlet air dry-bulb temperature to the temperature difference between the intake air dry-bulb and wet-bulb. It is determined as follows.

$$\varepsilon_{\mathrm{wb}} = \frac{t_{\mathrm{db,1}} - t_{\mathrm{db,2}}}{t_{\mathrm{db,1}} - t_{\mathrm{wb,1}}} \tag{1}$$

Dew-point effectiveness defined as the ratio of difference between intake and outlet air dry-bulb temperature to the difference between intake dry-bulb and dew-point temperature. The dew-point effectiveness is determined by the following equation:

$$\varepsilon_{\mathrm{dp}} = \frac{t_{\mathrm{db,1}} - t_{\mathrm{db,2}}}{t_{\mathrm{db,1}} - t_{\mathrm{dp,1}}} \tag{2}$$

2.2.2 Cooling capacity and energy efficiency ratio

When the REC uses outside air as the intake air, the cooling capacity of the cooler can be evaluated using the sensible cooling of intake air. It can be determined by the equation derived from ASHRAE Standard 143 [34].

$$Q = \frac{c_{\mathrm{p,a}} \rho_{\mathrm{a}} V_2 (t_{\mathrm{db,1}} - t_{\mathrm{db,2}})}{3.6} \tag{3}$$

The energy efficiency ratio (EER) of REC can be defined as the ratio of sensible cooling capacity to total power consumption of the cooler (in this case, the overall electricity consumption by fans and pump):

$$\mathrm{EER} = \frac{Q}{P} \tag{4}$$

2.2.3 Water evaporation rate

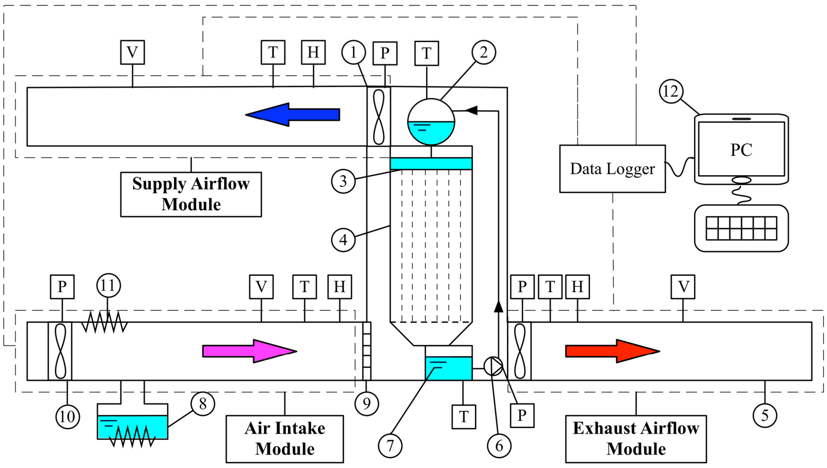
Water consumption of the REC depends on the airflow rate of working air in wet channels, the contaminant load of water, the effectiveness of the evaporator medium, and the dry-bulb and wet-bulb difference of the intake air. Ideally, the water evaporation rate is determined by taking the moisture rise from inlet to outlet of working air and multiplying by the working air mass flow rate divided by the density of water film. It can be calculated by the following formula:

$$V_{\mathrm{W}} = \frac{1000 V_3 \rho_3}{\rho_{\mathrm{W}}} (w_3 - w_1) \tag{5}$$

3. Experimental set-up

To evaluate the performance of the proposed counter-flow REC operating under various operation conditions, an experiment system was constructed as shown in Fig. 3. The experiment system consists of a REC heat and mass exchanger module, upper water reservoir and water distributor, bottom water reservoir, circulating water pump as well as the modules of intake, supply and exhaust air. A photograph of the experimental set-up in laboratory is shown in Fig. 4.

As indicated in Fig 3, the air intake module comprises a long extension duct and an inserted fan, filter and electrical heater and humidifier. The fan adjusted by variable speed controller that was used to vary the inlet airflow rate entering the REC module. The electric heater and humidifier regulate temperature and humidity of the air entering the REC. The supply air module includes an extension duct set up with a fan with variable speed controller to adjust the supply airflow rate of REC. In the same manner, the exhaust air module also involves a fan with variable speed controller mounted in a long extension duct to control the discharged exhaust flow rate. The ratio of exhaust to inlet airflow rate, i.e. working-to-intake air flow ratio, was varied by adjusting the supply and exhaust air fans simultaneously.



No	Components
1	Supply Air Fan
2	Upper Water Reservoir
3	Water Distributor
4	Heat Exchanger
5	Exhaust Air Fan
6	Circulating Water Pump
7	Bottom Water Reservoir
8	Electric Humidifier
9	Inlet Filter
10	Intake Air Fan
11	Electric Heater
12	Data Logging and Monitoring PC System

Measuring Parameters	
T	Temperature
H	Relative Humidity
V	Air Velocity
P	Electrical Power

Fig. 3 Schematic of experimental set-up of counter-flow regenerative evaporative cooler

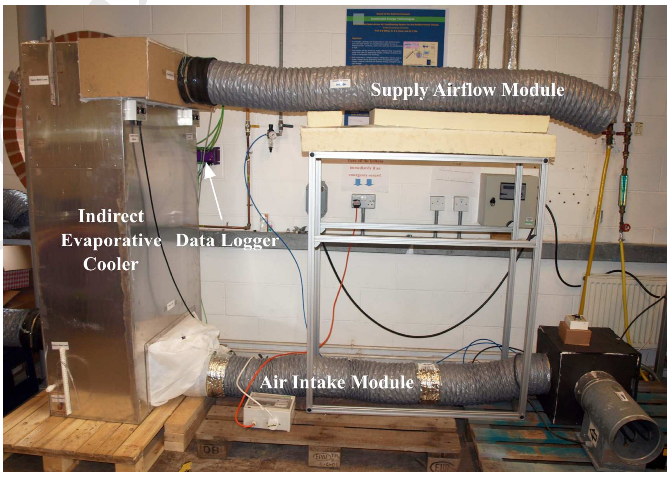


Fig. 4 Photograph of experimental set-up of counter-flow regenerative evaporative cooler

To achieve uniform wetting conditions for the wet channels of the REC, a circulating water system had been designed. The system involves upper and bottom reservoirs, water distributor, circulating water pipe and water pump associated with float switch. Water from household pipe could be used as good resources for wetting. The upper reservoir containing feed water was set above the top of HMX. From the reservoir, the water fell down to the connecting water distributors that were able to make water drip uniformly along the horizontal direction of wet channel surface. One part of the feed water had been absorbed owing to the wick structure of the inner surface of wet channel and then evaporated to the working air; The remaining water, being collected in the bottom reservoir, was pumped up to upper tank through a pipe for circulation.

3.1 Experimental apparatus and measurement method

Fig. 3 indicates the apparatus for measuring the experimental parameters, including air velocity, dry-bulb temperature, relative humidity of intake, supply and exhaust flows as well as power consumption. For the case of each tested flow, the velocity was measured using hotwire anemometer. The velocities at multiple points of the connecting duct were measured to obtain the arithmetic average velocity for the airflow. The temperature of airflow was measured by K-type thermocouples set up at various positions of connecting ducts to achieve average temperature of airflow. The relative humidity of air was measured by Vaisala HMP45A humidity probe with accuracy ±1% RH after calibration.

The water temperatures in the reservoirs were tested using PT100 resistance temperature sensors. An adjustable icy box was used to cool the feed water temperature, which was varied for the purpose of examining the effect of feed water temperature on system’s performance. The power consumption of the system was measured by power meter. Table 2 shows the measurement apparatus applied in this experiment study with corresponding measurement range and accuracy.

Table 2 Experimental apparatus applied in measuring parameters with range and accuracy of measurement

Parameters	Apparatus	Measurement range	Accuracy
air dry-bulb temperature	Type K thermocouples	-50 to +250 °C	±1.5 K or ±0.4%
air relative humidity	Humidity sensor	0-100% RH	±1% RH
air velocity	Hotwire anemometer	0-20 m/s	±0.03 m/s or 5%
water temperature	PT100 resistances temperature sensor	50 to +200 °C	±1.0 K
power consumption	Power meter	0-1000 W	±0.01 W

Nearly all measured data, except for the water level height and air velocities, were recorded at 5 s and transferred to a PC through a data logger (DT 500 Datataker). A computer programme was complied in the data logger for data acquisition and analysis. The measured data was analysed under both transient and steady states depending on the situations in which the cooler had been studied. The steady state was defined as the time when the temperature variation is within 0.1 ^{o}C and the relative humidity variation is within 1% for 15 min. When achieving steady state, the air temperature and humidity used for performance analysis were the value by considering the average of numerous measured data over a period of 10 min.

3.2. Uncertainty analysis

The uncertainty analysis for the measured results was performed to examine the validity of instrument output collectively as a whole. The results of concerned performance indicators were attained from the measuring variables indirectly. Eq. (6) obtaining from the reference [35] can be used to determine the relative uncertainty for the dependent performance indicator variables.

$$\frac{\Delta y}{y}=\left[\left(\frac{\partial f}{\partial x_1}\right)^2\left(\frac{\Delta x_1}{y}\right)^2+\left(\frac{\partial f}{\partial x_2}\right)^2\left(\frac{\Delta x_2}{y}\right)^2\cdots+\left(\frac{\partial f}{\partial x_n}\right)^2\left(\frac{\Delta x_n}{y}\right)^2\right]^{\frac{1}{2}} \tag{6}$$

Where Δy and $\Delta y/y$ represent the absolute and relative uncertainty of dependent variables respectively; f is the function of several independent variables, i.e. $x_1, \cdots, x_n$; Δx represents the absolute uncertainty of the independent variables.

All experimental data had been concerned for conducting the uncertainty check. The results of relative uncertainty for the performance indicator variables, as indicated in Table 3, were found to be within ±2.0% for wet-bulb effectiveness, ±5.3% for sensible cooling capacity, ±5.3% for EER, and ±5.7% for water evaporation rate.

Additionally, the accuracy of test results is also affected by the locations of measurement sensors. Particularly, the accuracy of airflow rate derived from air velocity measurements depends largely on uniformity of the air velocity distribution. Energy balance between the product and working air in the REC was examined to verify the accuracy of flow rate and sensors position. Specifically, the energy changes in product and working air were obtained from Eqs.(7) and (8). Fig. 5 displays the comparison of the energy changes for all the experimental data conducted in this study. The gap between them was found to be within ±5%.

Table 3 Results of uncertainty analysis for concerned performance indicators

($t_{1,db}$=36.5 °C, $t_{1,wb}$=28.5 °C, u_1=0.7 m/s, V_3/V_1=0.46)

Symbol	Performance indicators	Nominal value	Relative uncertainty
ε_{wb}	Wet-bulb effectiveness	1.05	±1.71%
ε_{dp}	Dew-point effectiveness	0.78	±1.92%
V_2	Product Airflow rate	120 (m^3/h)	±5.03%
Q	Cooling capacity	432 (W)	±5.29%
EER	Energy efficiency ratio	10.6	±5.30%
V_3	Working airflow rate	104 (m^3/h)	±5.02%
V_w	Water evaporation rate	0.88 (kg/h)	±5.69%

$$Q_p = m_1(h_1 - h_2) \tag{7}$$

$$Q_w = m_3(h_3 - h_2) \tag{8}$$

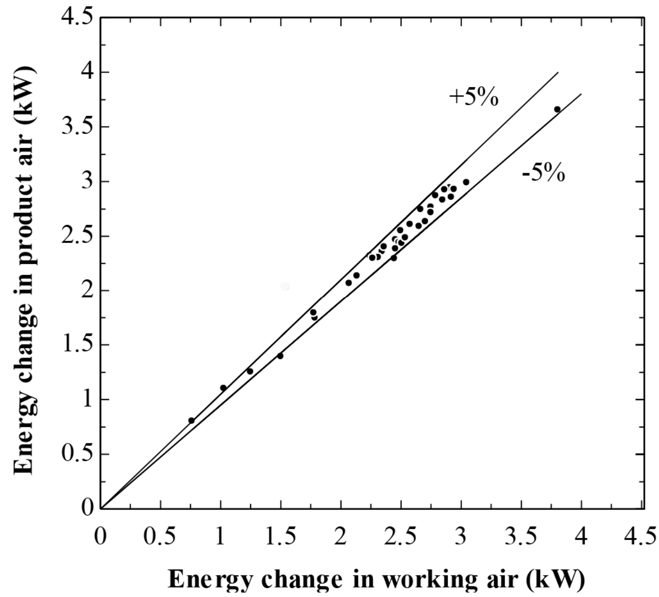


Fig. 5 Comparison between energy change in product and working air

4. Discussions of experiment results

4.1 Results for module performance

Table 5 presents the experimental results of performance for the REC unit operating at high and low speeds of fan under constant intake air dry-bulb temperature of 36 ^{o}C and wet-bulb temperature of 20 ^{o}C. The cooling capacity, cooling effectiveness and EER and were calculated based on the data of temperature, humidity and velocity measured for intake, outlet and exhaust air. It was found that the wet-bulb effectiveness of cooler ranged from 79 to 91%, the product air dry-bulb temperature varied from 21.4 to 23.4 ^{o}C with EER rating from 10-14.

Table 5 Experimental results for REC module performance ($t_{1,db}$=36 ^{o}C, $t_{1,wb}$=20 ^{o}C)

Symbol	Parameter	High Speed	Low Speed
$t_{2,db}$	Product air dry-bulb temperature	23.4	21.4
V_2	Product airflow (m^3/h)	257	120
u_1	Dry channel air velocity (m/s)	1.68	0.79
u_3	Wet channel air velocity (m/s)	0.84	0.39
V_3/V_1	Working-to-intake air ratio	0.5	0.5
P	Power consumption (W)	81	60
ε_{wb}	Wet-bulb effectiveness (%)	79	91
Q	Cooling capacity (W)	1133	608
EER	Energy efficiency ratio	14	10

4.2 Effect of inlet air wet-bulb depression

Keeping the intake channel air velocity constant at 1.58 m/s and working-to-intake air ratio stable at 0.5, the inlet wet-bulb depression of the cooler, i.e. the temperature difference between intake air dry-bulb and wet-bulb, was being continuously increased by turning up the input power of the electrical heater at the air intake of the experiment system. As a result, both inlet air dry-bulb and wet-bulb temperatures were both increased from 22.7 to 38.9 ^{o}C and 16.5 to 21.7 ^{o}C respectively. Fig. 6 shows the transient experimental data obtained for intake air, product air and exhaust air temperatures as continuously increasing the intake air dry-bulb and wet-bulb temperature over a period of time while keeping intake channel velocity and working-to-intake air ratio constant at 1.58 m/s and 0.5 respectively. The product air dry-bulb temperature was raised from 20.8 to 25.6 ^{o}C by 24% with increasing the inlet air wet-bulb depression from 6 to 16.7 ^{o}C approximately.

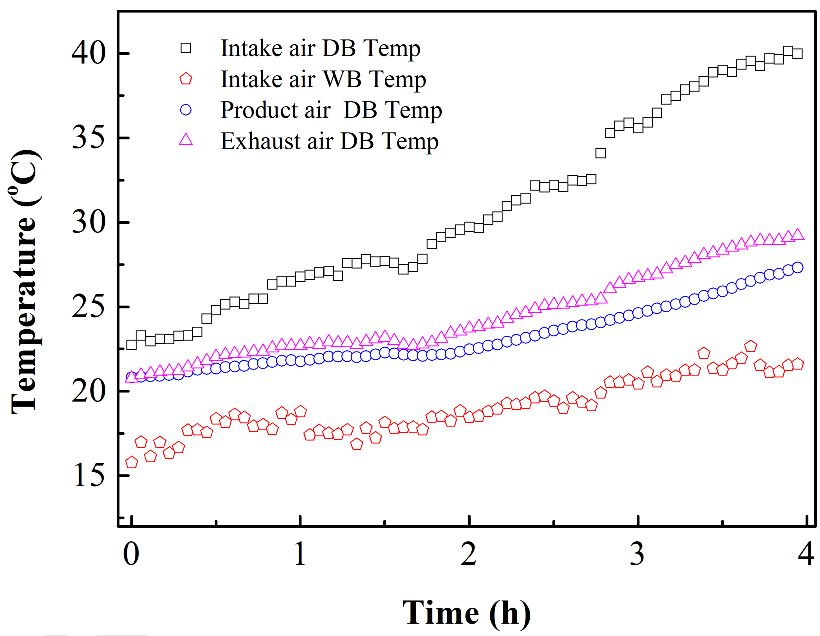
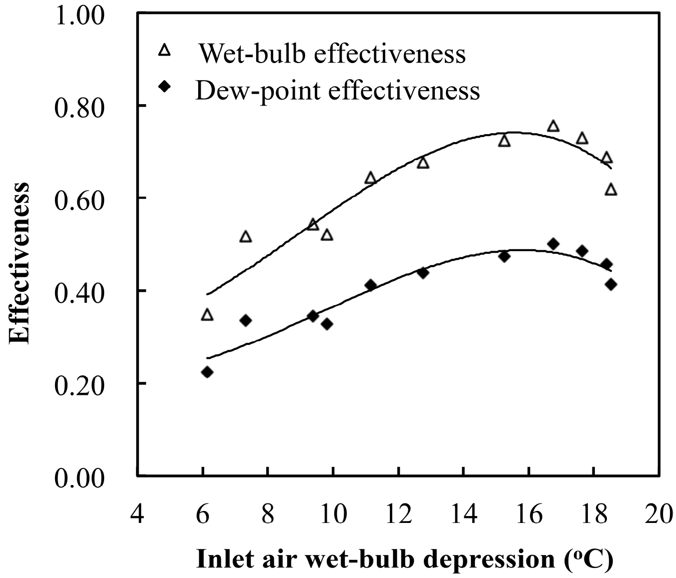


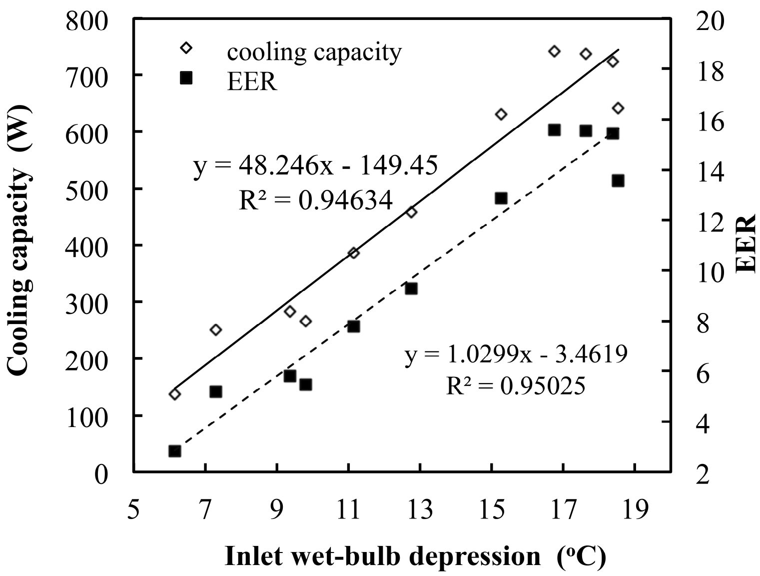
Fig. 6 Product and exhaust air temperature variations with increasing inlet dry-bulb depression (u_1=1.58 m/s, V_3/V_1=0.5)

The effectiveness, cooling capacity and EER of the presented cooler have been increased dramatically from 35% to 76%, 136 to 742 W and 2.8 to 15.5 respectively with the inlet wet-bulb depression increasing from 6 to 16.7 ^{o}C. The slight decreases were found for those parameters with the inlet wet-bulb depression increasing from

16.7 to 18.5 ^{o}C, as shown in Fig 7. Those decreases were caused by the condensation that occurred when the humidity and temperature of exhaust air was high enough. For



(a)



(b)

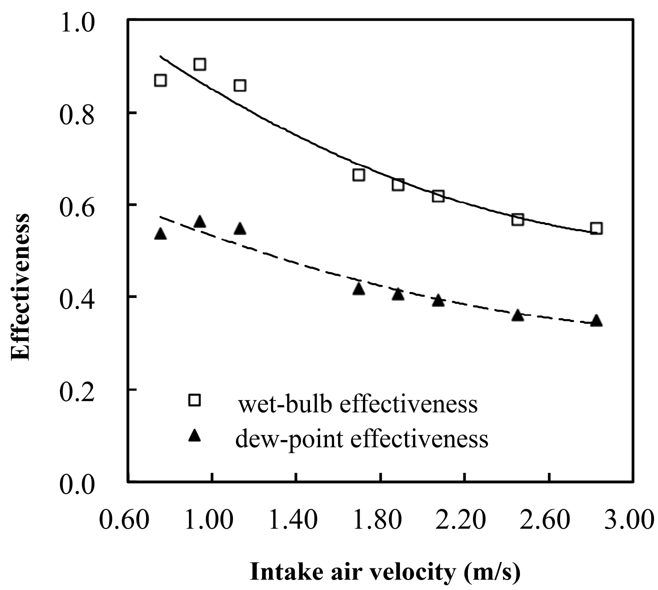
Fig. 7 Performance of testing system as a function of inlet wet-bulb depression (u_1=1.58 m/s, V_3/V_1=0.5) (a) effectiveness (b) cooling capacity and EER

this case, we tested the temperature and humidity of exhaust air at the place of the connecting extension duct. When exhaust air temperature is higher than the ambient room temperature, there would be a heat flux transferring from exhaust air to the ambient. The exhaust air temperature would be reduced and the condensation would occur. Therefore, the real temperature and humidity of exhaust air could not be measured correctly in this condition.

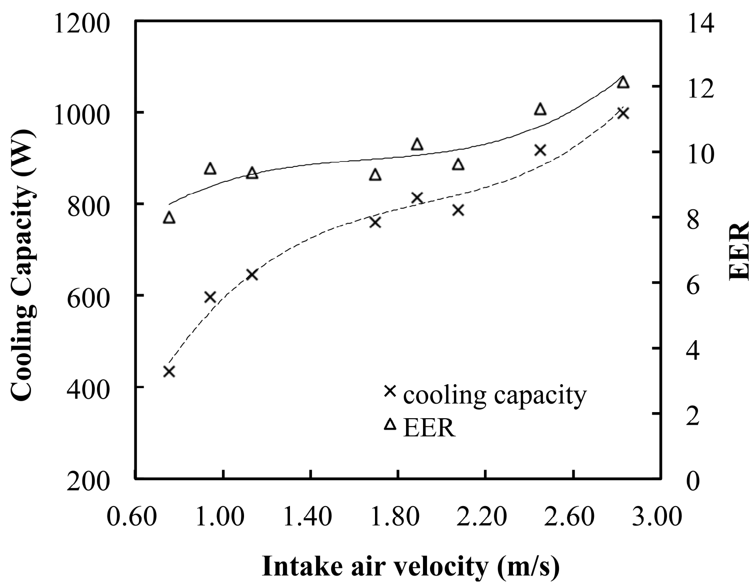
4.3 Effect of intake air velocity

Fig. 8 (a) is a graph obtained for the cooling effectiveness of the presented system versus air velocity of intake channel under constant working-to-intake air ratio of 0.5 and intake air temperatures of 36 ^{o}C dry-bulb and 20 ^{o}C wet-bulb. It was noticed that the wet-bulb and dew point effectiveness were dropped steadily by 35% and 31% respectively with increasing the intake air velocity from 0.75 to 2.83 m/s. The reason of this trend is due to the reduced contact time between working air and wet surface that give rise to decreased evaporative rate.

Fig. 8 (b) shows the measurement data of cooling capacity and EER of REC versus air velocity of intake channel. Both cooling capacity and EER were observed to increase from 433 to 1000 W and 8.0 to 12.1 respectively with increasing the intake air velocity from 0.75 to 2.83 m/s. The EER had gained a rise because the increase of cooling capacity was higher than that of power consumption.



(a)

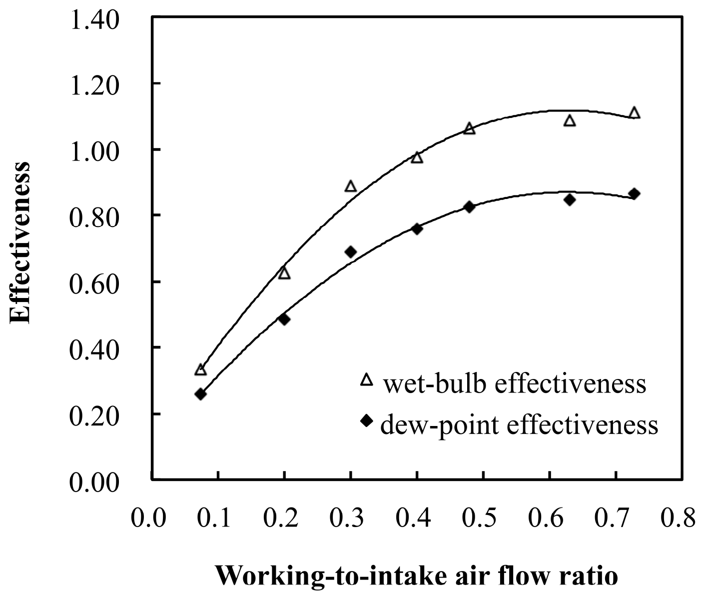


(b)

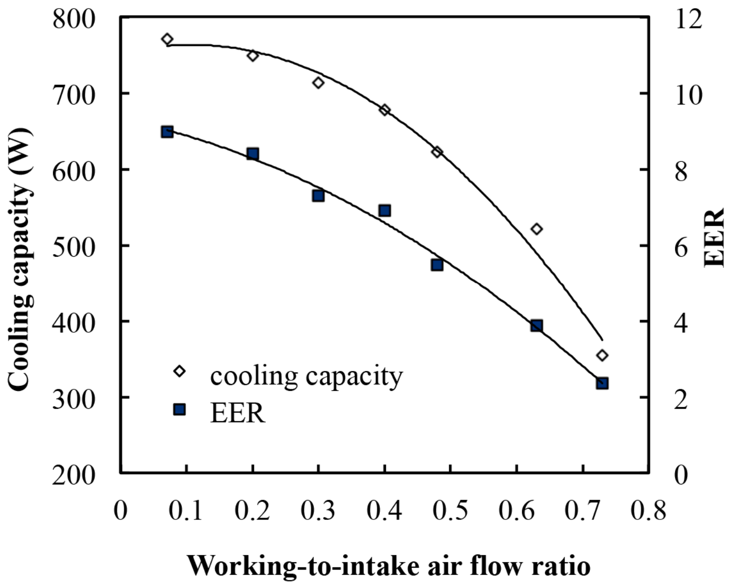
Fig. 8 Performance of testing system versus intake air velocity ($t_{1,db}$=36 ^{o}C, $t_{1,wb}$=20 ^{o}C, V_3/V_1=0.5) (a) effectiveness (b) cooling capacity and EER

4.4 Effect of working-to-intake air flow ratio

Fig. 9 (a) plots the test results obtained for cooling effectiveness of the REC versus working-to-intake air flow ratio under constant intake air temperatures of 36.5 ^{o}C dry bulb, 28.5 ^{o}C wet bulb and intake channel velocity of 0.83 m/s. The effectiveness was increased significantly from 0.26 to 1.06 with increasing the working-to-intake air flow ratio from 0.1 to 0.5. With the ratio increasing further from 0.5 to 0.7, the effectiveness was increased slightly from 1.06 to 1.11. This effect can be explained by Anisimov's numerical study [36]. When working air mass flow rate is smaller than primary, its heat capacity and ability to assimilate water vapour is relatively small. That is why primary air is cooled less effective. When working airflow rate is larger than primary air, its heat capacity allows for good assimilation of water vapour and sensible heat. Therefore, the cooling effectiveness is higher. Opposite trend was found in Fig. 9 (b). With increasing the working-to-intake air flow ratio, both cooling capacity and EER of the cooler were significantly decreased as a consequence of reduced supply airflow rate and increased working airflow rate.



(a)



(b)

Fig. 9 Performance of testing system versus working-to-intake air flow ratio ($t_{1,db}$=36.5 ^{o}C, $t_{1,wb}$=28.5 ^{o}C, u_1=0.83 m/s) (a) effectiveness (b) cooling capacity and EER

From the above analysis, the improvement in effectiveness could be achieved by the means of increasing working-to-intake air flow ratio but at the cost of reducing cooling capacity. Therefore, a reasonable working-to-intake air flow ratio would enable the cooler to reach a compromise between effectiveness and cooling capacity. Considering the slight increase in effectiveness with the working-to-intake air flow ratio increasing from 0.5 to 0.7, the appropriate ratio should be set in the range of 0.4-0.5.

4.5 Effect of water temperature

Fig. 10 shows the graph obtained for effectiveness of the cooling prototype operating at fixing intake air dry-bulb and wet-bulb temperatures, channel velocity and working-to-intake air ratio ($t_{1,db}$=36 ^{o}C, $t_{1,wb}$=20 ^{o}C, u_1=1.58 m/s, V_3/V_1=0.5) versus feed water temperature. The effectiveness was decreased by 11.8 % with the water temperature rising from 14.1 to 18.9 ^{o}C and further gradually reduced only by 5% as the temperature changing from 18.9 to 23.1 ^{o}C. Therefore, the negligible effect of feed water temperature is due to the latent heat transfer of water evaporation, which is much more important than sensible heat transfer in the process of evaporative cooling

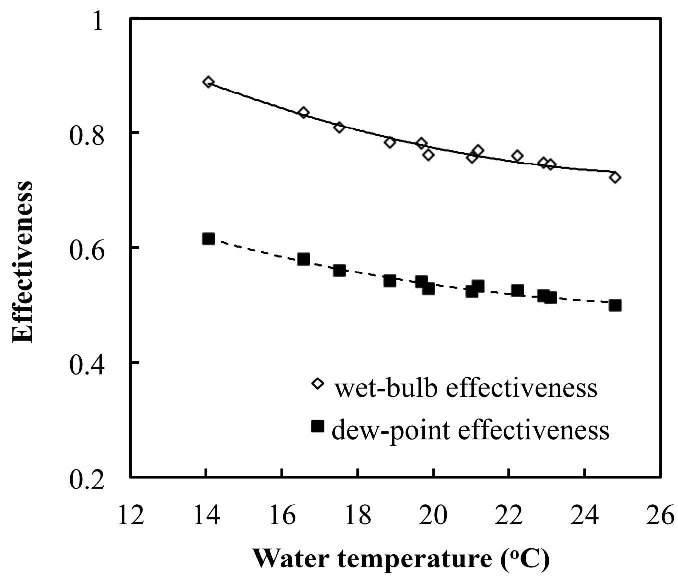


Fig. 10 Effectiveness versus feed water temperature ($t_{1,db}$=36 ^{o}C, $t_{1,wb}$=20 ^{o}C, u_1=1.58 m/s, V_3/V_1=0.5)

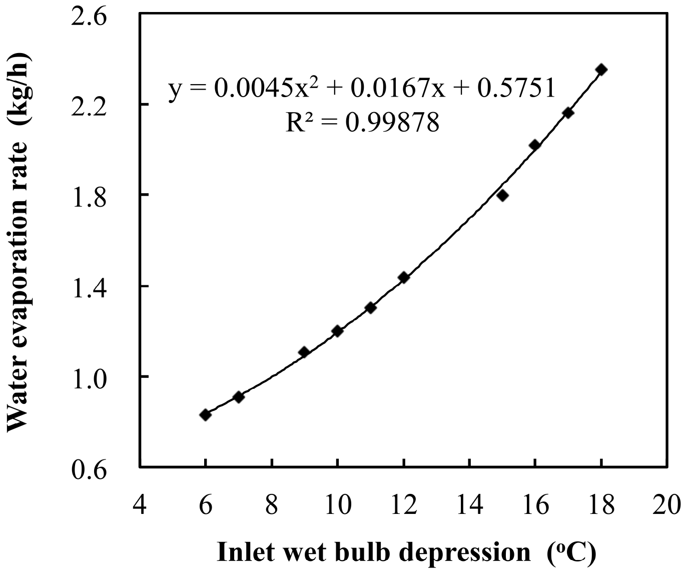
4.6 Water evaporation rate

Water evaporation rate of the REC unit can be determined by the amount of increase in the humidity ratio of working air multiplying by the working airflow rate during evaporation process. The real water consumption rates of the REC was difficult to acquire considering existing measurement errors, such as inaccurate readings of water level indicators and neglect of spraying or wasted water, etc. Therefore, the water evaporation rate was obtained using the measurement data of working air humidity ratio and volumetric flow rate.

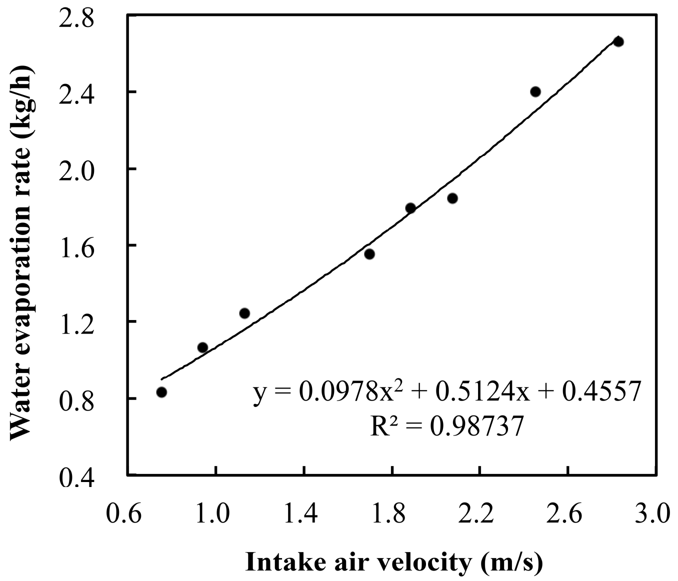
As depicted in Fig. 11(a), the water evaporation rate was enhanced from 0.83 to 2.35 kg/h with increasing the inlet wet-bulb depression under constant working-to-intake air flow ratio of 0.5 and intake channel velocity of 2.2 m/s. This can be explained by the reason that the difference of vapour pressure between wet surface and working air would be increased with the increase of inlet wet-bulb depression. Therefore, the evaporation rate would be enhanced consequently. The experiment data of water evaporation rate versus inlet wet-bulb depression was well fitted by a quadratic polynomial correlation, as indicated in Eq.(9). It is useful in the optimal design of feed water flow rate and would only be used in the certain operating condition specified in Fig.11(a).

$$V_{\mathrm{w}} = 0.0045(t_{1,\mathrm{db}} - t_{1,\mathrm{wb}})^2 + 0.0167(t_{1,\mathrm{db}} - t_{1,\mathrm{wb}}) + 0.5751 \tag{9}$$

Fig. 11(b) shows that the water evaporation rate was enhanced significantly from 0.83 to 2.66 kg/h as the intake air velocity increased from 0.75 to 2.83 m/s under constant working-to-intake air flow ratio of 0.5 and intake air dry-bulb and wet-bulb temperatures of 36 ^{o}C and 20 ^{o}C. The upward trend can be explained in this manner: the working air velocity, passing through the internal wet surface of heat and mass exchanger, was increased as the result of increasing intake air velocity. The rising speed of working airflow would accelerate the evaporation rate of water film.



(a)



(b)

Fig. 11 Water evaporation rate as a function of intake air conditions (a) inlet wet-bulb depression (u_1=2.2 m/s, V_3/V_1=0.5) (b) intake air velocity ($t_{1,db}$=36 ^oC, $t_{1,wb}$=20 ^oC, V_3/V_1=0.5)

4.7 Comparison between experimental and numerical results

The experimental data obtained from the measurements were compared with the verified computer model from our published work [14]. The impact of the corrugated air guiders on the performance of cooler was taken into account in the numerical modelling. The air guiders are in the shape of corrugation that blocked around 25% of heat transfer area of the exchanger, therefore the actual heat transfer area was reduced by 25%. A computer modelling, considering the reduction in heat transfer area, was conducted to achieve the numerical effectiveness of the proposed REC. The comparison between the numerical and experimental data was made on the basis of the identical geometry and operating conditions of the cooler. Table 6 shows the comparative results of wet-bulb effectiveness, product and exhaust air temperatures for different inlet air dry-bulb and wet-bulb temperatures. The discrepancy was found to be within 4.7% between numerical and experimental results. Apart from the factor of air guider blocking, other factors, such as incomplete wetting conditions of wet surface and uniformed air distribution in dry and wet channels could lead to the undesirable reduction in the effectiveness of the prototype.

Table 6 Comparison between numerical and experimental wet-bulb effectiveness for different inlet air DB and WB Temp (u_1=0.83 m/s, V_3/V_1=0.4)

$T_{1,db}$	$T_{1,wb}$	$T_{2,db}$		ε_{wb}		Discrepancy ($T_{2,db}$, ε_{wb})	$T_{3,db}$		Discrepancy ($T_{3,db}$)
^oC	^oC	Exp, ^oC	Sim, ^oC	Exp	Sim	%	Exp, ^oC	Sim, ^oC	%
24.0	17.3	19.2	18.4	0.800	0.837	4.4	21.2	20.3	4.2
25.7	17.5	18.9	18.8	0.840	0.844	0.5	21.2	21.0	0.9
26.3	18.2	19.8	19.4	0.830	0.849	2.3	22.2	21.7	2.3
30.6	19.7	21.7	21.1	0.846	0.871	2.9	24.8	24.1	2.8
34.2	21.9	24.3	23.2	0.850	0.891	4.6	27.8	26.5	4.7
34.7	23.6	25.4	24.7	0.872	0.898	2.9	28.4	27.5	3.2
34.8	22.9	24.8	24.1	0.870	0.896	2.9	28.1	27.2	3.2

4.8 Comparison with previous studies

Performance and parameters of the REC developed in this study has been compared with those of the REC in previous studies, as shown in Table 7. It was found that the present laboratory unit had similar wet-bulb effectiveness and higher energy efficiency compared with the REC products tested by Bruno's [24] and Elberling's [23], but the cooling capacity was lower than those of products. Compared with Riangvilaikul and Kumar's [21] as well as Jradi and Riffat's [28] work, the present cooler had a slight lower effectiveness and higher energy efficiency due to the larger channel spacing or smaller channel length.

Table 7 Comparison of regenerative evaporative coolers

Parameters	Present study	Ref [21]	Ref [24]	Ref [28]	Ref [23]
Study method	Exp	Exp/sim	Exp	Exp/sim	Exp
Flow arrangement	Counter flow	Counter flow	Counter flow	M-cycle cross flow	M-cycle cross flow
Channel spacing (dry/wet), mm	6/6	5/5	N/A	5/5	N/A
Channel length, m	0.9	1.2	N/A	0.5	N/A
Intake air temperature, ^{o}C	22.7-38.9	25-45	22.5-40.3	26.1-41.1	26.7-43.8
Intake air humidity, g/kg	9.3-9.4	7-26	1.07-26.7	3.1-19.0	9.6-10.7
Intake air velocity, m/s	1.58-2.83	2.4	N/A	2	N/A
Working to intake air ratio	0.5	0.33	N/A	0.33	0.5
Product air temperature, ^{o}C	20.8-25.6	15.6-32.1	14.7-21.8	22.2-29.2	19.9-25.6
Wet-bulb effectiveness	0.55-1.06	0.92-1.14	0.93-1.06	0.77-1.17	0.81-0.91
Cooling capacity, kW	0.14-1.13	N/A	15-20	1.05-1.22	9.18-12.17
EER	2.8-15.5	N/A	4-11.5	6.8-14.2	9.3-9.6

5. Conclusion

A regenerative evaporative cooler was fabricated in laboratory and the cooling performance of the cooler has been investigated experimentally under various operating conditions. In addition, this study investigated the impacts of operating parameters on the overall performance. The proposed regenerative cooler is mainly composed of a counter-flow polygonal-sheets-stacked plate heat and mass exchanger consisting of multiple groups of dry and wet channels inserted with triangular air guider. The heat transfer plate of the exchanger was made of aluminium and the internal surface of wet channels was coated with porous fibre layer to enhance surface wettability. The acquired experimental results were compared using our published numerical model and the reasons for discrepancy have been analysed.

For the present experimental study, a few specific conclusions can be drawn: The wet-bulb effectiveness of the present REC module varied from 0.55 to 1.06 with the EER ranged from 2.8 to 15.5 under various operating conditions. The cooling effectiveness of the presented REC module was significantly improved with increasing intake wet-bulb depressions, reducing intake air velocity, or increasing working-to-intake air ratio respectively. The cooling capacity and EER of the cooler were improved through increasing intake wet-bulb depressions or intake air velocity, or alternatively by reducing the ratio of working to intake airflow rate. To achieve high effectiveness, we should allow intake air velocity to be small enough, however, this would reduce the cooler's cooling output, which may not satisfy the cooling requirement for spaces. The working-to-intake air ratio choosing from 0.4 to 0.5 is appropriate in order to achieve a compromise between effectiveness, cooling capacity and energy efficiency. The feed water temperature has a negligible impact on performance of the REC. The water evaporation rate of the cooler can be greatly improved by increasing the inlet wet-bulb depression or the intake air velocity provided that the working-to-intake air flow ratio is fixed.

Acknowledgement

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Highlights

- A counter-flow regenerative evaporative cooler (REC) was constructed and tested.
- Performance of REC was evaluated experimentally.
- Effects of operating parameters on REC's performance were found.
- Effects of intake air conditions on water evaporation rate of REC were explored.
- The experiment data obtained was compared with numerical results.