THE ENERGY AND EXERGY ANALYSIS OF COUNTER-FLOW REGENERATIVE EVAPORATIVE COOLER

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Recently the regenerative evaporative cooler (REC) has drawn great attention from researchers because it can cool the intake air below the wet-bulb temperature and approaching its dew point temperature. For further understanding of the heat and mass transfer occurred in a counter-flow REC, a novel mathematical model is developed based on the law of energy conservation and the principle of the thermodynamic theory. The proposed mathematical model is validated against experimental data from literature. The parametric study is performed to investigate the performance of the REC under different operating and geometrical conditions. It is found that the exergy destruction and exergy efficiency ratio of the REC are strongly influenced by the intake air velocity, the working to intake air ratio and channel gap, followed by the channel length. The working to intake air ratio choosing from 0.3 to 0.4 is appropriate in order to achieve better thermal performance with permissible level of thermodynamic cost. Moreover, the results obtained in this paper reveal that the best thermal performance does not correspond to the best thermodynamic performance. Thus, both the first and second law of thermodynamics should be considered for a comprehensive analysis.

Key words: *regenerative evaporative cooler, heat and mass transfer, energy, exergy*

1. Introduction

With the advent of the vapor compression cycle, the air conditioning market has been dominated by vapour compression refrigeration systems. This cooling mode, owing to the usage of the power intensive compressor and associated chemical refrigerants, e.g. HCFC, is neither sustainable nor environmentally friendly. Without the participations of compressor and refrigerant, evaporative cooling is considered to be a potential substitute for mechanical compression system. However, the conventional evaporative cooling has very limited temperature reduction potential because it cannot lower the product air temperature to the wet bulb temperature of intake air.

Recently, the regenerative evaporative cooler (REC), as an advanced evaporative cooling configuration, has attracted great attention for providing the air below the wet bulb temperature of intake air without moisture content increase. Zhao et al. [1] presented numerical simulation of a counter-flow heat and mass exchanger that features the triangular cross section of air channels. The numerical simulation indicated that the cooling effectiveness and energy efficiency are dramatically affected by the dimensions of the air flow passages, air velocity and the working-to-intake-air ratio, and less dependent on the temperature of the feed water. Riangvilaikul and Kumar [2] carried out numerical studies of dew point evaporative cooling system with counter-flow configuration under a variety of intake air conditions. They purposed the optimum parameters of the counter-flow REC and investigated the effectiveness. Hasan [3] developed an analytical model for the counter-flow REC based on modified ε-NTU method. The outlet product air temperature had around 7.4% discrepancy compared to experimental data. Pandelidis et al. [4] numerically investigated the performance of typical REC and REC with perforations based on a modified ε-NTU model. It was found that the working to intake air ratio had a significant impact on the cooling performance of both coolers. Duan et al. [5] presented an experimental study of a counter-flow REC with triangular air guider under various operational conditions. The results revealed that the wet-bulb effectiveness of the REC rated from 0.55 to 1.06 with Energy Efficiency Ratio varied from 2.8 to 15.5. Kabeel and Abdelgaied [6] carried out experimental and numerical studies of a novel counter-flow REC with internal baffles in the dry channel. They demonstrated that the wet-bulb effectiveness of the novel REC increased with increased number of baffles. Moshari and Heidarinejad [7] presented a numerical investigation into the cooling performance comparison of cross flow and counter flow REC. It was found that the counter-flow REC could offer greater cooling capacity. Cui et al. [8] developed a simple performance correlation for counter-flow REC and the correlation was further validated with experimental data. The results showed the dimensionless product air outlet temperature was within a discrepancy of 12%. Duan et al. [9] carried out experimental and numerical studies of a prosed large scale counter-flow REC under a variety of intake air conditions, and the wet-bulb effectiveness in the range of 96%-107% could be achieved. Boukhanouf et al. [10] proposed a validated numerical model and conducted experimental study of the REC using porous ceramic materials. It was found that the cooling capacity of the wet surface area approched 225 W/m^2 and the wet-bulb effectiveness of 102% could be achieved.

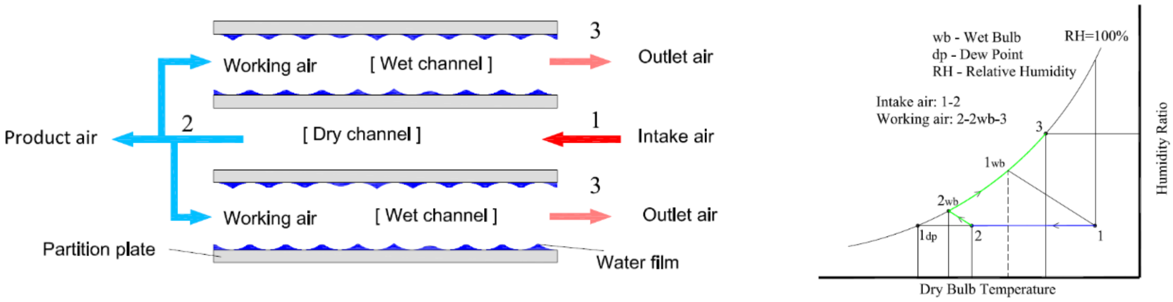
In summary, the heat and mass transfer characteristics and performance of the counter-flow REC have been evaluated by the first law of thermodynamics through numerical simulations and experiments methods. However, the first law method alone is not adequately able to show some important viewpoints. It neglects the existence of the energy quality and irreversibility of the thermodynamic process. On the contrary, the exergy analysis, known as the second law method, can characterize the irreversibility of the heat and mass transfer processes within the REC and fulfill of the incompleteness. The exergy analysis method has been widely applied in the performance evaluation of various energy systems, including evaporative systems [11-13].

As mentioned above, the previous research mainly focused on energy analysis and few researchers consider the applicability of exergy analysis for the counter-flow REC investigation. Therefore, to overcome the shortfall in the theoretical study of the REC , the paper has presented an energy and exergy analysis using both the first and second laws of thermodynamics for the counter-flow REC. By using the experimentally validated model, a parametric study is conducted based on the performance parameters of the REC including cooling capacity, dew point effectiveness, exergy destruction and exergy efficiency ratio under different operational and geometrical conditions. The work of the study is expected to help researchers better understand the thermal and thermodynamic characteristics of the REC and provides some original information for the improving design.

2. Mathematical model for counter-flow REC

2.1. Description of counter-flow REC

A counter-flow REC as illustrated in Fig. 1(a), normally comprises of numerous fibrous plates that are stacked together. One side of each plate is applied with a water-proof coating, thus each forming a dry surface (coating side) and wet surface (uncoated side) for the channels of the REC. Dry surfaces of two adjacent plates are against each other to form numerous dry channels; whilst the wet surfaces of the two adjacent plates are also against each other to form numerous wet channels.



(a) (b)

Figure 1. Principle of regenerative evaporative cooling: (a) Schematic of counter flow REC (b) psychrometric process of counter flow REC

During operation, the intake air flows in the dry channel and part of its outlet flow is employed as the working air, which flows in the adjacent wet channel in reverse direction. The plate's surface of wet channel is wetted by water, resulting in the sensible heat transfer caused by temperature difference and mass transfer caused by water vapor pressure difference between the wet surface and working air. The intake air is cooled without contacting water directly, thus humidity ratio remains unchanged. The psychrometric process of counter flow REC is showen in Fig. 1(b).

2.2. Heat and mass transfer model

Normally, a counter-flow REC consists of numerous dry channels and wet channels separated from each other by partition plate. Therefore, a representative cell element of the REC, which comprises half the gap of a dry channel, a plate and half the gap of a wet channel, is selected as shown in Fig. 2. The model is employed to each individual element, and then the temperature and moisture distribution within the cell element can be obtained. The following reasonable assumptions have been adopted to simplify the developed model: The air is deemed to be incompressible; The process is steady state; The outer surface is thermally isolated from the surroundings; The physical properties of the air, water and vapor are assumed to be constant; The heat and mass transfer coefficient of moist air conforms to the Lewis relation; The water film is distributed uniformly across the wet channel, and the thermal resistance of water film and the thermal conductivity of the plate are neglected [14].

Fig. 2 shows the cell control element of the counter-flow REC. Based on the above assumptions, the energy balance in the intake air stream can be written as:

$$dq = -\frac{m_{dry}}{2} c_{p,a} dt_p = h_{dry}(t_p - t_w)dA \tag{1}$$

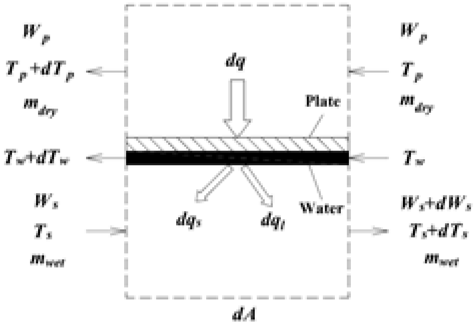


Figure 2. cell control element applied for calculation

The energy balance equation for the working air and water film of wet channel is expressed as

$$\frac{m_{wet}}{2} di_s = -[h_{wet}(t_w - t_s) + i_v h_m (W_{w,sat} - W_s)]dA \tag{2}$$

The mass balance equation for the working air and water film of wet channel can be described by the following equations,

$$\frac{m_{wet}}{2} dW_s = -h_m (W_{w,sat} - W_s)dA \tag{3}$$

$$\frac{m_{wet}}{2} dW_s = -dm_w \tag{4}$$

The convective mass transfer coefficient is expressed as a function of the convective heat transfer coefficient and the Lewis number [15], and the Lewis number is assumed to be equal to one:

$$\frac{h}{h_m} = \rho c Le^{2/3} \tag{5}$$

Due to the relatively small channel size and low velocity in dry and wet channels, the air streams can be treated as laminar flow. The thermal entry length for the laminar airflow in the channel can be calculated from [16],

$$l_{th} = 0.05 D_h \,\mathrm{RePr} \tag{6}$$

The Nusselt number [17] in the entry region can be calculated using the following empirical correlation

$$\mathrm{Nu} = \frac{\dfrac{7.54}{\tanh\left(2.264 Gz_D^{-1/3} + 1.7 Gz_D^{-2/3}\right)} + 0.0499 Gz_D \tanh\left(Gz_D^{-1}\right)}{\tanh\left(2.432 Pr^{1/6} Gz_D^{-1/6}\right)} \tag{7}$$

$$Gz_D = \frac{D_h}{x} \mathrm{RePr} \tag{8}$$

For a fully developed laminar flow, the Nusselt number [17] is constant as following

$$\mathrm{Nu} = 8.235 \tag{9}$$

The sensible heat transfer in the wet channel is expressed as follows

$$dq_{sen} = h_{wet}(t_w - t_s)dA = -\frac{m_{wet}}{2}c_{s,a}dt_s \quad (10)$$

The energy balance equation for the cell control element is given as

$$\frac{m_{dry}}{2}c_{p,a}dt_p + \frac{m_{wet}}{2}di_s + m_w c_w dt_w + c_w t_w dm_w = 0 \quad (11)$$

Eqs. (1)-(3) and (10)-(11) are a set of equations describing the heat and mass transfer process in the counter-flow REC. The Newton iteration method was adopted to solve these coupled discrete equations in Engineering Equation Solver environment. The grid independence test was performed under the specified conditions with different numbers of cell elements. The trial computation results showed that the change of product air outlet temperature is within 0.04 ℃ with the cell number increasing from 30 to 50. Therefore, by taking both accuracy of the solution and computing time into consideration, thirty cell control elements were distributed along the air flow direction.

The dew point effectiveness is an important index to evaluate the cooling performance of the REC, and the mathematical expression can be written as

$$\eta_{dp} = \frac{t_{p,db,in} - t_{p,db,out}}{t_{p,db,in} - t_{p,dp,in}} \quad (12)$$

Besides, the product air is cooled at the constant moisture content, the cooling capacity can be calculated as following

$$Q = m_p c_{p,a}(t_{p,db,in} - t_{p,db,out}) = m_{dry}(1-\theta)c_{p,a}(t_{p,db,in} - t_{p,db,out}) \quad (13)$$

The local resistances of air flow and power consumption of the pump in the system are neglected. The theoretical required fan power of the system is obtained as following

$$p = \Delta P_{dry}V_{dry} + \Delta P_{wet}V_{wet} \quad (14)$$

2.3. Numerical validation

By solving the coupled equations stated above, the air temperature and humidity ratio of each cell control element were calculated to obtain the performance of the counter-flow REC. The model was further validated with published experimental data from [18].

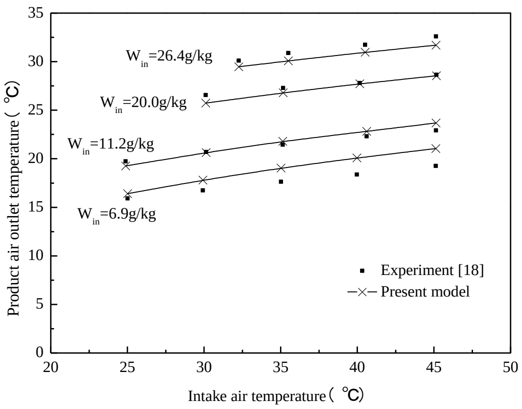


Figure 3. The numerical results compared with experimental data

The present model was set to the same operational/ geometrical conditions as the experiment data obtained from [18]. The physical sizes(channel length, channel width, channel gap) of the counter-flow REC were 1200 mm, 80 mm,and 5 mm, respectively. The experiment study was performed for intake air velocity of 2.4 m/s and working to intake air ratio of 0.33. Fig. 3 shows the comparison between the experiment data and numerical results. When the humidity ratio of intake air is 11.2 g/kg, 20.0 g/kg, 26.4 g/kg, the errors of product air outlet temperature is small. As intake air temperature increases when humidity ratio is 6.9 g/kg, the error becomes bigger and the biggest error is 9.7%. Taking into account the accuracy of test equipment and relevant assumptions, the established model can be used to simulate the heat transfer and mass transfer process in counter-flow REC.

3. Calculation method of exergy

Exergy is defined as the maximum work that can be obtained from a given form of energy using the environmental parameters as the reference state. The results of exergy analysis for a climatisation system strongly depend on the selection of the reference state [19]. Usually, a steady-atmospheric state is selected as the reference environment. However, when the atmospheric air is unsaturated, it still has available energy [20]. Therefore, the saturated condition of intake air is defined as the dead state for humid air and water.

The exergy in the air-conditioning process reaching thermal, mechanical and chemical equilibriums with the atmospheric environment can be written as

$$e_t = (i - i_0) - T_0(s - s_0) + \sum_{k=1}^{n} x_k(\mu_{k,0} - \mu_{k,00}) \tag{15}$$

Where the first two terms on the right side of Eq. (15) are the thermomechanical exergy, and the last term is the chemical exergy.

The moist air and water are the only two kinds of fluids involving in the REC. The moist air can be considered to be a mixture of ideal gases composed of dry air and water vapor. The exergy of moist air and water can be written as [12]

$$\begin{aligned} e_a = (c_{da} + Wc_v)T_0\left[\frac{T}{T_0} - 1 - \ln\frac{T}{T_0}\right] + (1+1.608W)R_aT_0\ln\frac{P}{P_0} + \\ R_aT_0[(1+1.608W)\ln\frac{(1+1.608W_{00})}{(1+1.608W)} + 1.608W\ln\frac{W}{W_{00}}] \end{aligned} \tag{16}$$

$$e_w = \left(i_w - i_{w,0}\right) - T_0\left(s_w - s_{w,0}\right) + \upsilon_w\left(P_w - P_{w,0}\right) - R_vT_0\ln\varphi_0 \tag{17}$$

The first two terms on the right side of Eq. (16) are the thermomechanical exergy of moist air, and the last term is the chemical exergy. In Eq. (17) the first three terms represent the thermomechanical exergy, and the last term represents the chemical exergy of the water.

The process is adiabatic with no work delivered, so the exergy balance for the REC represented by Fig. 1(a) is calculated as follows,

$$\left(m_1e_{1,a} + m_{w,in}e_{w,in}\right) = \left(m_pe_{2,a} + m_3e_{3,a}\right) + I \tag{18}$$

Where the terms in brackets on the left side of Eq. (18) are the total exergy entering into the REC; whereas, the first term on the right side represents the exergy leaving the REC, and the last term represents the destroyed exergy.

In the process of heat and mass transfer, added moisture to air is considered as an input quantity of water. Equations (16) and (17) are substituted into Eq. (18) to acquire the exergy destruction. For free energy involving in the service system, the exergy efficiency ratio is defined as an important index to evaluate the effective use of the purchased available energy. As the exergy efficiency ratio is higher , the REC is more profitable. The exergy efficiency ratio is defined as:

$$EER_{ex} = \frac{m_p e_{1,a,th}}{\left(m_1 e_{1,a,me} + m_3 e_{3,a,me}\right)} = \frac{m_p (1-\theta) e_{1,a,th}}{\left(m_1 e_{1,a,me} + m_3 e_{3,a,me}\right)} \tag{19}$$

Where the numerator of Eq. (19) is the thermal exergy of product air, and the denominator represent the mechanical exergy of the intake air and working air.

4. Simulation results and analyses

According to the above model, the numerical simulations have been undertaken to investigate the impact of the selected operational and geometrical parameters on the thermal performance (cooling capacity and dew point effectiveness) and thermodynamic performance (exergy destruction and exergy efficiency ratio) obtained by the counter-flow REC. The dimensions of the counter-flow REC are set initially to 1.0 × 1.0 × 0.1 m^3. The start-up conditions for simulation applied to the study are listed in Tab. 1.

Table 1. Pre-set structural and operational conditions for simulation

Channel length (mm)	Channel width (mm)	Channel gap (mm)	Wall thickness (mm)	Intake air temperature (°C)	Intake air humidity ratio(g/kg)	Intake air velociy (m/s)	Working to intake air ratio
1000	100	5	0.4	33.5	8.244	1.5	0.33

4.1. Effect of working to intake air ratio

Influence of working to intake air ratio on cooling capacity, exergy destruction, dew point effectiveness and exergy efficiency ratio is analyzed under different working to intake air ratio spanning 0.1 to 0.9 and the results are shown in Fig. 4 (a) and Fig. 4 (b). As shown in Fig. 4, when the working to intake air ratio is varied from 0.1 to 0.9, the variation trend of cooling capacity is similar with that of exergy efficiency ratio. The cooling capacity and exergy efficiency ratio are almost direct proportional to the working to intake air ratio firstly till 0.3. After this, the cooling capacity and exergy efficiency ratio are inversely proportional to the working to intake air ratio. That is because both the mass flow rate of product air and the temperature drop have a coupled impact on the cooling capacity and exergy efficiency ratio. Increasing in working to intake air ratio can lead to the reduction of product air to the cooling space. The exergy destruction and dew point effectiveness increase at the same time with working to intake air ratio rising. It is due to the fact that an increase in working to intake air ratio results in the enhancement of heat and mass transfer in the wet channel. Therefore, the temperature drop between product air inlet and outlet and exergy destruction both increase, as the amount of pressure drop is almost same. As the working to intake air ratio rises, the exergy destruction is increased by 1.7 times from 47.9 W to 130 W, and the exergy efficiency ratio is decreased by 85% from 21.3 to 3.2. Moreover, the working to intake air ratio choosing from 0.3 to 0.4 is reasonable enable the cooler to reach a compromise between thermal performance and thermodynamic performance.

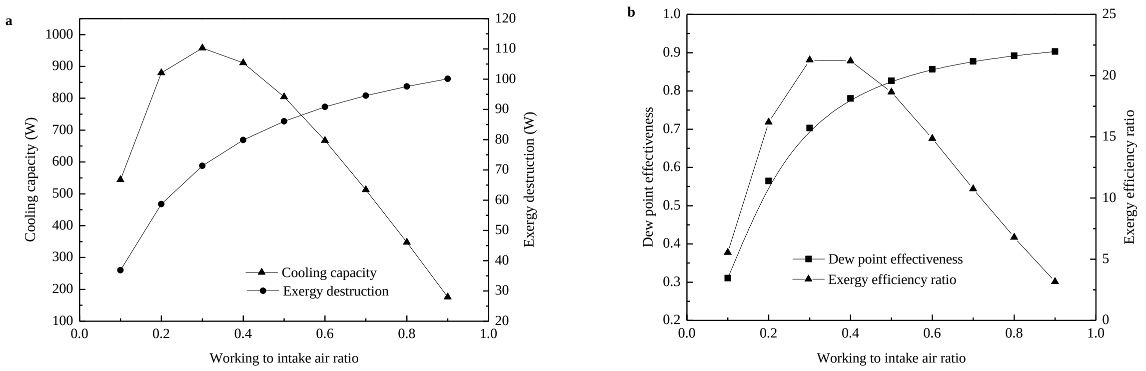


Figure 4. Influence of working to intake air ratio: (a) cooling capacity and exergy destruction (b)dew point effectiveness and exergy efficiency ratio

4.2. Effect of intake air velocity

The impact of intake air velocity on the performance of the REC is studied, shown in Fig. 5(a) and Fig. 5(b). Fig. 5(a) shows the cooling capacity and exergy destruction depending on the intake air velocity varying from 0.4 to 4.0 m/s. It can be observed that the cooling capacity and exergy destruction quickly increases with increasing the intake air velocity. Fig. 5(b) presents the dew point effectiveness and exergy efficiency ratio depending on the intake air velocity. The figure shows the dew point effectiveness and exergy efficiency ratio decrease at the same time with intake air velocity rising. That is because the increase of intake air velocity can lead to the increase of intake air mass flow rates in both channels. The increase of intake air velocity is relatively big comparing to the decrease of the product air outlet temperature. In addition, the higher intake air velocity can lead to the higher pressure loss. Thus, the mechanical exergy is bigger in both channels while the thermal exergy of the product air outlet is samller. It is concluded that both energy and exergy analyses should be considered for the optimization of the process to get the best performance. As the intake air velocity rises from 0.4 to 4.0 m/s, the exergy destruction is increased by about 8.5 times from 25.8W to 245.1W, and the exergy efficiency ratio is reduced by about 96.5%, from 123.5 to 4.3. These indicate that the effect of intake air velocity is greater than the working to intake air ratio on the exergy destruction and exergy efficiency ratio.

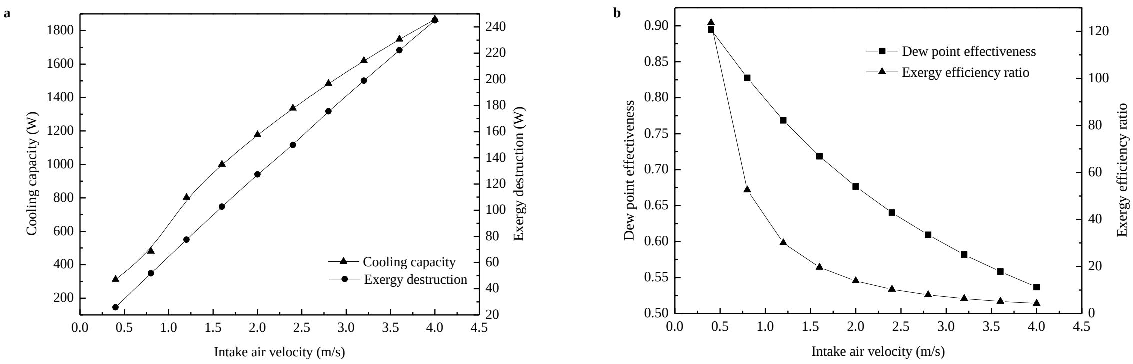


Figure 5. Effect of intake **air velocity: (a) cooling capacity and exergy destruction (b) dew point effectiveness and exergy efficiency ratio**

4.3. Effect of channel length

Simulations are performed to investigate the effect of channel length on cooling capacity, exergy destruction, dew point effectiveness and exergy efficiency ratio, and the results are illustrated in Fig. 6(a) and Fig. 6(b). It can be seen from Fig. 6(a) that the cooling capacity and exergy destruction increase with varying the channel length from 0.25 to 3.5 m. The rates of cooling capacity and exergy destruction increase slow down when channel length exceeds 1.5 m. The Fig. 6(b) presents the dew point effectiveness and exergy efficiency ratio depending on the channel length. By comparison, the changing trend of dew point effectiveness is similar with that of cooling capacity but is opposite with that of exergy efficiency ratio. That is because the increase of the length can increase the residence time and contact area, which is conducive to heat and mass transfer process. However, the increased channel length means increased pressure loss and mechanical exergy. Consequently, the exergy destruction increases from 85 W to 100.4 W, increased by about 0.18 times, as the value of channel length rises. Meanwhile, the exergy efficiency ratio decreases from 33.2 to 9.3, decreased by about 0.72 times. Therefore, the effect of channel length is smaller than the working to intake air ratio on the performance of exergy destruction and exergy efficiency ratio. It is also concluded that that the best thermal performance does not correspond to the best thermodynamic performance.

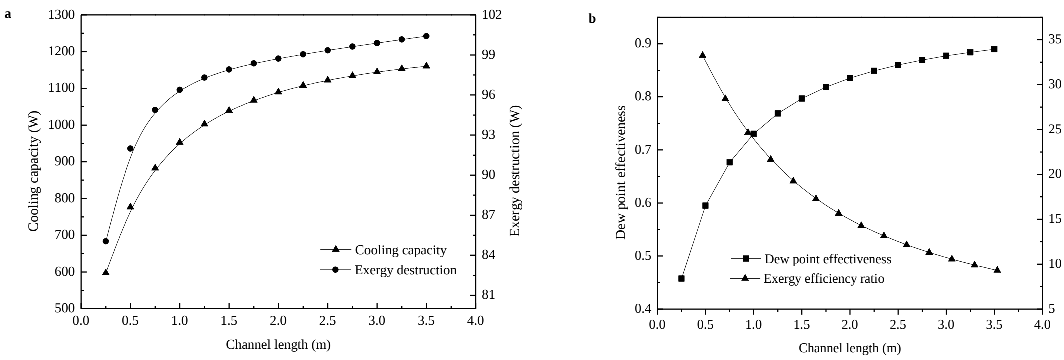


Figure 6. Effect of channel length: (a) cooling capacity and exergy destruction (b) dew point effectiveness and exergy efficiency ratio

4.4. Effect of channel gap

To investigate the impact of channel gap on the thermal performance and thermodynamic performance of the REC through changing the channel gap from 2mm to 12mm while other parameters keep unchanged under the pre-set conditions and the results are presented in Fig. 7(a) and Fig. 7(b). As shown in Fig. 7(a), the cooling capacity and exergy destruction quickly decreases with increasing the channel gap. Fig. 7(b) indicates that the exergy efficiency ratio increases but the dew point effectiveness decreases almost linearly with channel gap rising. This can be attributed to a high channel gap resulting in decreased flow resistance. The mechanical exergy is smaller in both channels while the thermal exergy of the product air outlet is bigger. The decrease of mechanical exergy is relatively small comparing to the increase of the product air outlet temperature. It should be noted that the small channel gap can provide good thermal performance but bad thermodynamic performance. As the channel gap decreases, the exergy destruction is increased by about 27% from 80.8 W to 102.9 W, while the exergy efficiency ratio is decreased by about 81% from 31.6 to 5.9. Results come out that the effect of channel gap is smaller than the working to intake air ratio on the exergy destruction and exergy efficiency ratio, yet it is greater than the effect of the value of channel length.

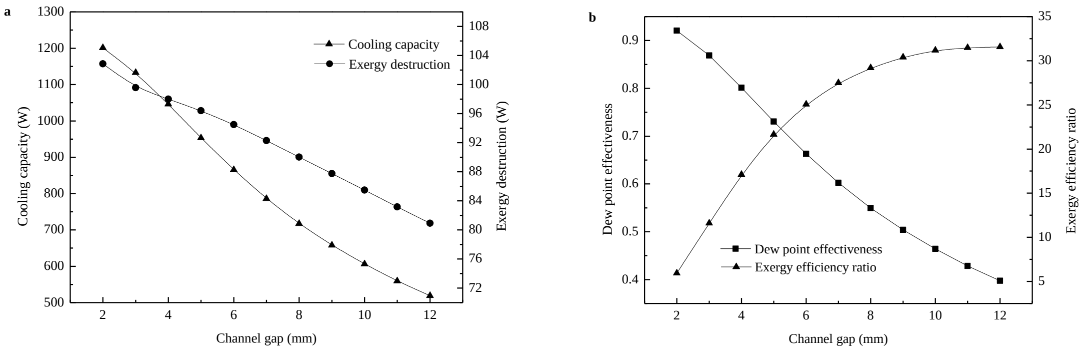


Figure 7. Effect of channel gap: (a) cooling capacity and exergy destruction (b) dew point effectiveness and exergy efficiency ratio

5. Conclusions

This study prensents a mathematical model for the counter-flow REC based on the heat and mass transfer theory. The model was validated by the experimental results from the literature. An exergy balance is derived in order to determine the exergy destruction associated with heat and mass transfer. The performance of the considered REC is parametrically evaluated under various operating and geometrical conditions in terms of cooling capacity, exergy destruction, dew point effectiveness and exergy efficiency ratio. The main conclusions can be drawn from the present study as follows:

(1) The working to intake air ratio choosing from 0.3 to 0.4 is appropriate in order to achieve a compromise between thermal performance and thermodynamic performance. To achieve good thermodynamic performance, the intake air velocity should be small enough, however, this would reduce the cooling capacity.

(2) The exergy destruction is increasing with increasing the working to intake air ratio, intake air velocity and channel length, or reducing channel gap respectively, while the exergy efficiency ratio is opposite to that. Furthermore, it is found that the influence of operational and structural parameters on the exergy destruction and exergy efficiency ratio from large to small are listed as follows: intake air velocity, working to intake air ratio, channel gap, channel length.

(3) The big channel length or small channel gap can provide large cooling capacity and great dew point effectiveness, however, the exergy destruction is reduced with increasing channel length but decreasing channel gap. The operational and geometrical parameters have complex effects on thermal performance and thermodynamic performance of the counter-flow REC. The results show that the optimum situation for obtaining thermal performance can not match with the best thermodynamic performance. Thus, the energy and exergy analysis should be implemented simultaneously for the optimization of the process to get the better thermal performance at permissible level of thermodynamic cost.

For a future work, exergetic analysis and comparison of this kind of counter-flow REC with conventional air cooling systems will be developed.

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Nomenclature

A	heat and mass transfer area cell control element [m^2]	**Subscripts**	
c	specific heat capacity [$Jkg^{-1}K^{-1}$]	a	air
D_h	hydraulic diameter [m]	da	dry air
e	specific exergy [Jkg^{-1}]	db	dry bulb
EER_{ex}	exergy efficiency ratio	dp	dew point
Gz_D	Graetz numbe	dry	dry channel
h	heat transfer coefficient, [$Wm^{-2}K^{-1}$]	in	inlet
h_m	mass transfer coefficient [m/s]	lat	latent
i	enthalpy [J/kg]	me	mechanical
l	length [m]	out	outlet
Le	Lewis number	p	proruct air
m	mass flow rate [kg/s]	s	working air
I	exergy destruction [W]	sat	saturation state
Nu	Nusselt number	sen	sensible
p	theoretical fan power [W]	t	total
P	pressure [Pa]	th	thermal
$\triangle P$	pressure loss [Pa]	υ	specific volume [m^3/kg]
Pr	Prandtl number	v	water vapor
q	heat transfer rate [W]	w	water
Q	cooling capacity [W]	wet	wet channel
R	gas constant, [$Jmol^{-1}K^{-1}$]	0	restricted dead state
Re	Reynolds number	00	dead state
T	thermodynamic temperature [K]		
t	Celsius temperature [°C]	**Greek symbols**	
V	air volume flow rate [m^3/s]	η	effectiveness
W	humidity ratio, kg moisture/kg dry air	ρ	density [kg/m^3]
x	mole fraction	φ	relative humidity
μ	chemical potential [J/kg]	θ	working to intake air ratio

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