THE ENERGY AND EXERGY ANALYSIS ON THE PERFORMANCE OF COUNTER-FLOW HEAT AND MASS EXCHANGER FOR M-CYCLE INDIRECT EVAPORATIVE COOLING

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The dew point Indirect Evaporative Cooling (IEC) achieved through Maisotsenko cycle (M-Cycle) is a complicated thermodynamic process. For further understanding of the heat and mass transfer occurred in a dew point indirect evaporative air cooler with M-Cycle counter-flow configuration, the paper presents a novel mathematical model that combined the law of energy conservation and the principle of the thermodynamic theory. The model was used to carry out the parametric study of the dew point air cooler under various inlet air temperature and relative humidity. Through the combined analysis of energy and exergy of the target IEC system, it is found that both the inlet air temperature and relative humidity have an important effect on the thermal performance and thermodynamic performance of the heat and mass exchanger. The high temperature environment helps to get better thermal performance and thermodynamic performance. It has been showed in this paper that the best thermal performance does not correspond to the best thermodynamic performance. Thus, the energy and exergy analysis should be implemented simultaneously for the optimization of the process to get the best thermal performance at permissible level of thermodynamic cost.

Key words*: indirect evaporative cooling, heat and mass transfer, energy, exergy*

1. Introduction

Evaporative cooling is a technology that makes use of the air and evaporation of the spread water to perform the cooling for the air. In general, evaporative cooling can be classified into the two categories: direct evaporative cooling (DEC) and indirect evaporative cooling (IEC). For DEC, the process air is cooled with contacting water directly, thus moisture content increase and the dry air is turned into moist air. For IEC, the temperature of product air can be cooled without increasing humidity that makes IEC more attractive than DEC. However, the minimum temperature of supply air that can be reached by both DEC and traditional IEC is confined to the wet bulb temperature of inlet air.

A new type of heat and mass exchanger (HMX) taking advantage of the Maisotsenko cycle (M-Cycle) [1] has attracted great attention in recent years for providing the air below the wet bulb temperature of inlet air without moisture content increase. Usually, the heat and mass transfer characteristics and performance of the HMX are investigated through numerical simulations [2-6], experiments [7-8] and analytic methods [9-11]. However, the most of the previous studies made use of the mathematical models that were sourced from the first law of thermodynamics and neglected the existence of the energy quality and irreversibility of the thermodynamic process. Nevertheless, the exergy analysis, known as the second law method, can characterize the irreversibility of the heat and mass transfer processes within the HMX and fulfill of the incompleteness of the energy analysis alone. In conjunction with exergy analysis, the energy applied can be utilized better and HMX design can be oriented towards a possible state of thermodynamic perfection.

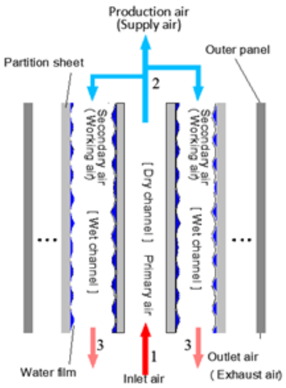
Few researchers have made efforts to study on the exergy analysis of this type of HMX, but over the past few years, a lot of studies have been carried out on the evaporative systems assisted with exergy and energy analysis. Bejan [12] derived the exergy expressions for moist air and water, and investigated the thermal performance of an adiabatic evaporative cooling process by exergy analysis. Lu *et al*. [13] performed the exergy analysis on evaporative condenser and cooling tower using an ozonation system. It was shown that the exergy analysis could provide better clarity of explanation for performance than the energy analysis. Ren *et al*. [14] evaluated the thermodynamics performance of four schemes of evaporative cooling by selecting saturated outdoor air as dead state. The results showed that the regenerative evaporative cooling had the highest profit and the wet bulb effectiveness had a significant impact on improving the exergy efficiency. Muangnoi *et al*. [15] developed a validated model for studying the exergy characteristic of air and water through the counter flow cooling tower height. The results revealed that the evaporative heat transfer plays a dominated role in exergy of moist air and the exergy destruction is highest at the bottom of the cooling tower. Furthermore, they [16] carried out the exergy analysis of the cooling tower in various inlet air conditions and the fixed water condition based on the validated numerical model. Santos *et al*. [17] presented numerical simulation of the air washers under a variety of inlet air conditions, water temperature and air washer length. The numerical simulation indicated that the best thermal comfort for the intake air is not agreement with the best condition for thermodynamic performance. Qureshi and Zubair [18] conducted exergetic analysis on the thermal performance of counter flow cooling towers and evaporative cooling heat exchangers, and the results showed that the exergy efficiency of all the systems increased with the inlet wet bulb temperature rising. Taufiq *et al*. [19] carried out the exergy analysis of the direct evaporative cooling system in a Malaysian building, demonstrating that the method was well suited for analyzing energy consumption. Saravanan et al. [20] theoretically performed the energy and exergy analysis on cooling tower using an ozonation system. From the results showed that the inlet air wet bulb temperature was the most important parameter through the analysis. Moien [21] studied the exergy analysis of the direct, indirect and direct/indirect evaporative cooling based on the experimental results for six cities in Iran respectively. The results obtained showed that the cities of Bam, Tehran and Yazd as well as their associated climates respectively used direct, indirect and direct/indirect evaporative cooling reasonably. Wang and Li [22] proposed a validated numerical model and conducted a parametric study of the counter flow cooling tower with the second law analysis. It was found that the exergy parameters were susceptible to the cooling efficiency when it is firmly near 1.0 at lower water to air rates. Hakan Caliskan *et al*. [23] compared and analyzed the exergetic performance of three traditional air cooling system and the novel M-Cycle system under twelve different reference temperatures ranging from -5 ℃ to 50 ℃. The obtained results showed that exergy efficiency of the novel system was highest when the reference temperature was higher than 23 ℃. Zhang *et al*. [24] defined a novel parameter unmatched coefficient for exergy analysis between water and humid air. It was concluded that exergy efficiency raised and exergy destruction decreased when the parameter was close to 0.

Accordingly, to overcome the shortfall in the theoretical study of this type of exchanger and to further enable optimization of the exchanger performance, the paper has presented an exergy analysis for the dew point counter-flow IEC that was built based on the principle of the exergy theory and mass & energy conservation theories, thus leading to better evaluate the thermodynamic performance of the HMX for dew point IEC. Based on this model, some key characterization parameters of energy efficiency and thermodynamic perfection, such as cooling capacity, dew point effectiveness, exergy efficiency, and so on, have been investigated under different operational conditions. The results of the study will be of significant importance in terms of better understanding of the energy and exergy transfer processes inside and improving cooling performance and energy efficiency of such apparatus.

2. Mathematical model for counter-flow HMX

2.1. Description of counter-flow HMX

A dew point counter-flow heat and mass exchanger (HMX) of IEC, which is designed on the M-Cycle principle as illustrated in Fig. 1, normally comprises of numerous fibrous sheets that are stacked together with the support of the paralleled studs. One side of each sheet 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 HMX. Dry surfaces of two adjacent sheets are opposed to form numerous dry channels separated by the air guides; whilst the wet surfaces of the two adjacent sheets are also opposed to form numerous wet channels separated by the air guides.



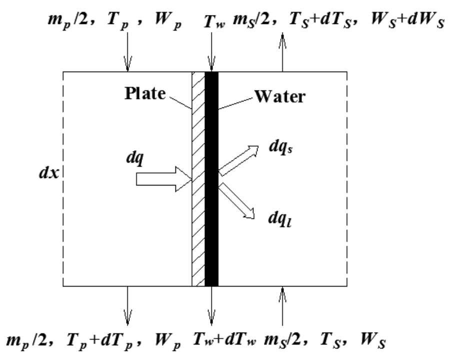


Figure 1. Schematic of counter flow HMX **Figure 2. cell control element applied for calculation**

During operation, the primary air flows in the dry channel and part of its outlet flow is employed as the secondary 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 secondary air. The primary air is cooled without contacting water directly, thus humidity ratio remains unchanged.

2.2. Heat and mass transfer model

Normally, a counter-flow HMX of dew point IEC consists of numerous dry channels and wet channels separated from each other by partition panels. Therefore, a representative cell element of the exchanger, which comprises half the height of a dry channel, a plate and half the height of a wet channel, is selected as shown in Fig. 2. The model was employed to each individual element, and then the temperature and moisture distribution within the cell element can be obtained. The following reasonable assumptions were made to simplify the developed model:The flow is incompressible and in steady state;The outer surface is thermally isolated from the surroundings; The heat and mass transfer coefficient of moist air conforms to the Lewis relation, and the Lewis number is assumed to be equal to one. 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 [25].

Fig. 2 shows the cell control element of the counter-flow dew point IEC. Based on the above assumptions, the energy balance in the dry channel can be written as：

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

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

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

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

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

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

The convective mass transfer coefficient is expressed as a function of the convective heat transfer coefficient and the Lewis number [26]:

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

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 [27],

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

The Nusselt number [28] 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 [28] is constant as following

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

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

$$dq_{sen} = h_s(t_w - t_s)dA = -\frac{m_s}{2}c_{s,a}dt_s \tag{10}$$

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

$$\frac{m_p}{2}c_{p,a}dt_p + \frac{m_s}{2}di_s + m_w c_w dt_w + c_w t_w dm_w = 0 \tag{11}$$

Eqs. (1)-(3) and (10)-(11) are a set of equations describing the heat and mass transfer process in the counter-flow HMX of dew point IEC.

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

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

Besides, the primary air is cooled at the constant moisture content, the cooling capacity can be expressed as follows

$$Q = m_f c_{p,a}(t_{p,db,in} - t_{p,db,out}) \tag{13}$$

The local resistances of air flow and power consumption of the pump in the system are neglected. The theoretical fan power of the system can be expressed as

$$p = \Delta P_p V_p + \Delta P_s V_s \tag{14}$$

The Newton Raphson iteration method was employed to solve these coupled discrete equations in Engineering Equation Solver (EES) environment. The grid independence test was performed under the specified conditions with the cell number increasing from 30 to 50. The trial computation results showed that the change of supply air temperature is within 0.04 ℃. Taking into consideration both accuracy of the solution and computing time, thirty cell control elements were distributed along the air flow direction.

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 HMX of dew point IEC. The model was further validated with published experimental data from [29].

Woods and Kozubal [29] carried out an experiment study on a counter flow HMX of dew point IEC and the temperature of primary outlet air was measured. The physical sizes(channel length, channel width, channel height of dry channel and wet channel) of the HMX were 500 mm, 500 mm, 1.85 mm and 2 mm, respectively. Tab. 1 presents the experimental data compared with numerical results under the different inlet air conditions. It is found that the maximum discrepancy between the numerical results and experimental data for the outlet temperature of primary air is about 0.93℃ and the maximum relative error is 6.3%. Through the above analysis, it is apparent that the numerical results agree well with the experimental data and the present model is able to provide a reasonable accuracy at predicting the thermal performance of the dew point IEC.

Table 1. The numerical results compared with experimental data

Test #	m_p (kg/s)	m_s (kg/s)	$t_{p,in}$ (°C)	$W_{p,in}$ (kg/kg)	$t_{p,out}$ (°C)		Discrepancy
					experiment	simulation	%
1	0.154	0.046	35	0.0034	9.7	9.65	-0.5
2	0.154	0.031	35	0.0096	18	18.2	1.1
3	0.154	0.015	35	0.0096	24.6	25.5	3.7
4	0.108	0.032	35	0.0095	14.2	13.8	-2.8
5	0.154	0.046	35	0.0162	19.3	20	3.6
6	0.154	0.046	28.9	0.0097	14	14.5	3.6
7	0.154	0.046	35	0.0099	14.49	15.24	5.2
8	0.154	0.046	43.3	0.0098	14.85	15.78	6.3
9	0.153	0.015	25	0.0107	21.01	21.18	0.8

3. Calculation method of exergy

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

$$e_t = e_{th} + e_{me} + e_{ch} \tag{15}$$

The moist air and water are the only two kinds of fluids involving in the HMX of dew point IEC. 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

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

$$e_w = \left[i_w(T) - i_w(T_0)\right] - T_0\left[s_w(T) - s_w(T_0)\right] - R_v T_0 \ln\varphi_0 \tag{17}$$

The selection and definition of the dead state condition is very important for exergy analysis. Usually, a steady-atmospheric state is selected as the reference environment. However, when the atmospheric air is unsaturated, it still has available energy. Such a humid air process will allow the water diffuse into the unsaturated air spontaneously and ultimately reach the saturated state. Therefore, the dead state is defined as saturated outdoor air in this paper.

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

$$\left(m_1 e_{1,a} + m_{w,i} e_{w,i}\right) = \left(m_f e_{2,a} + m_3 e_{3,a}\right) + I \tag{18}$$

In the process of heat and mass transfer, added moisture to air is considered as an input quantity of water. Eqs. (16) and (17) is substituted into Eq. (18) to acquire the exergy destruction. The second law exergy is an important index to evaluate the cooling performance, the rational exergy efficiency is defined as:

$$\eta_e = 1 - \frac{I}{\left(m_1 e_{1,a} + m_{w,i} e_{w,i}\right)} \tag{19}$$

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. The exergy efficiency ratio is defined as:

$$EER_{ex} = \frac{m_f e_{1,a,th}}{\left(m_1 e_{1,a,me} + m_3 e_{3,a,me}\right)} \tag{20}$$

4. Simulation results and analyses

Employing the above model, the performance of a counter flow HMX of dew point IEC were numerically investigated in this section. The dimensions of the counter flow HMX are set initially to $1.0\times1.0\times0.1$ m^3. The start-up conditions for simulation applied to the study are listed in Tab. 2.

Table 2. Pre-set structural and operational conditions for simulation

Channel length (mm)	Channel width (mm)	Channel gap (mm)	Wall thickness (mm)	Inlet air temperature (°C)	Inlet air humidity ratio(g/kg)	Inlet air velociy (m/s)	Working to intake air ratio
1000	100	5	0.5	33.5	8.244	1.5	0.33

4.1. Exergy transfer in the HMX

The simulation was begun under the nominal conditions, as shown in Tab. 3. For these nominal conditions, the simulation results are presented in Table 3. It can be seen from Tab. 3 that the thermal exergy of inlet air decreases but the chemical exergy remains constant. That is because the product air is cooled with keeping humidity ratio unchanged. In the wet channel, the thermal exergy and chemical exergy of secondary air are both decreasing. It is due to the fact that the dry bulb temperature and humidity ratio of secondary air are both increasing during heat and mass transfer process drawing near the dead state. Through the process, the thermal exergy and chemical exergy contained in the secondary air are partly removed to the product air but the change of chemical exergy is higher than that of thermal exergy. The mechanical exergy of air both lessens due to the flow resistance in the dry and wet channel. The schematic diagram of the primary and secondary air exergy analysis is showed in Fig. 3. Due to the small pressure drop, the change of mechanical exergy is very small. Thus, the EER value (21.6) is greater than one. The total exergy of the primary and secondary air still reduces on account of the irreversibility occurring in heat and mass transfer process.

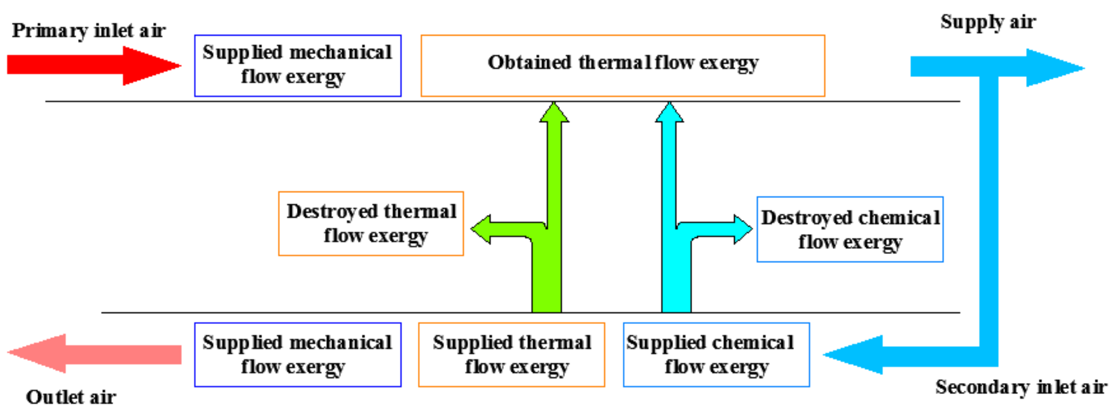


Figure 3. The schematic diagram of exergy analysis

Table 3. Simulation results under the nominal conditions

Air state	Thermal exergy (J/ kg)	Mechanical exergy (J/ kg)	Chemical exergy (J/ kg)	Total exergy (J/ kg)	η_e	EER_{ex}
Primary air inlet	0	12.9	1868	1880.9	59.2%	21.6
Supply air	459.9	0	1868	2327.9		
Secondary air inlet	459.9	4	1868	2331.9		
Outlet air	62.4	0	207.1	269.5		

4.2. Effect of inlet air temperature

Simulations are performed to investigate the influence of inlet air temperature on the thermal performance and thermal performance of the HMX through changing the inlet air temperature from 25 -45 ℃ while other parameters keep unchanged under the pre-set conditions and the results are presented in Fig. 4(a) and Fig. 4(b).

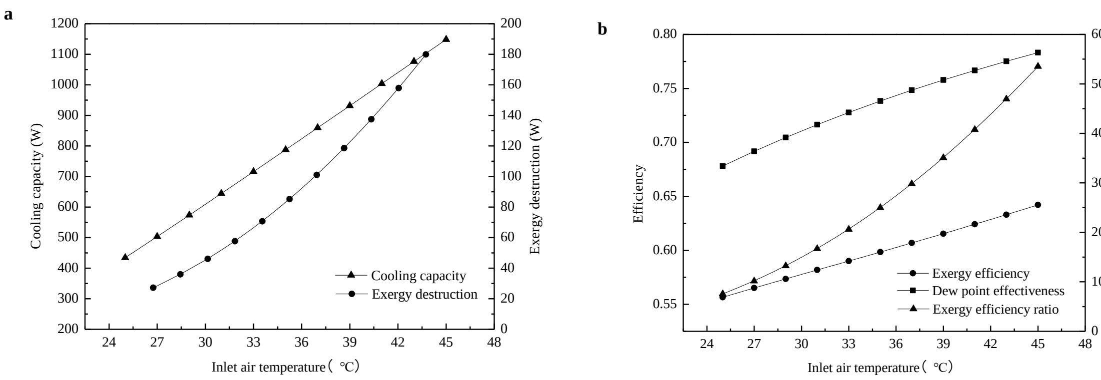


Figure 4. Influence of inlet air temperature: (a) cooling capacity and exergy destruction (b) exergy efficiency, dew point effectiveness and exergy efficiency ratio

Fig. 4(a) shows the influence of inlet air temperature on cooling capacity and exergy destruction. As is seen in Fig. 4(a), the cooling capacity and exergy destruction quickly increase with varying the inlet air temperature from 25-45 ℃. That is because the higher inlet air temperature raises the temperature difference between the air and water. Thus, the higher inlet air temperature results in the larger driving potential for heat exchange between the air and water and more sensible heat is removed from the air to help water evaporation. As a result, the temperature drop between inlet air and supply air increases and the irreversibility is high. The influence of inlet air temperature on exergy efficiency, dew point effectiveness and exergy efficiency ratio is illustrated in Fig. 4(b). The figure shows the exergy efficiency, dew point effectiveness and exergy efficiency ratio increase as increasing the inlet air temperature. When the inlet air temperature is high, the chemical exergy of inlet air is bigger and more thermal exergy and chemical exergy contained in the secondary air are partly removed to the product air, as the mechanical energy is almost unchanged. It can be observed that the greatest exergy efficiency, dew point effectiveness and exergy efficiency ratio occur when the cooling capacity is maximum. It is conclude that the high temperature environment helps to get better thermal performance and thermodynamic performance.

4.3. Effect of inlet air relative humidity

To investigate the impact of inlet air relative humidity on the thermal performance and thermodynamic performance of the HMX, the inlet air relative humidity is changed from 0.1-0.9, while other parameters keep unchanged under the pre-set conditions. The results are presented in Fig. 5(a) and Fig. 5(b).

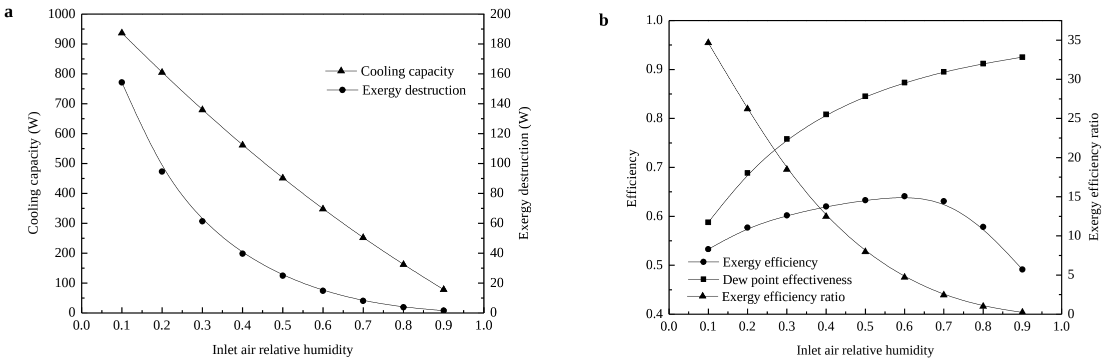


Figure 5. Influence of inlet air temperature: (a) cooling capacity and exergy destruction (b) exergy efficiency, dew point effectiveness and exergy efficiency ratio

Fig. 5(a) depicts the impact of inlet air relative humidity on cooling capacity and exergy destruction. It can be seen that the cooling capacity and exergy destruction quickly decrease with varying the inlet air relative humidity from 0.1-0.9. When the relative humidity is low, the driving force for mass transfer caused by water vapor pressure difference between the wet surface and secondary air is greater. Thus, this increases the chance for heat and mass transfer and the exergy destruction is high. The impact of inlet air relative humidity on exergy efficiency, dew point effectiveness and exergy efficiency ratio is shown in Fig. 5(b). The figure shows the exergy efficiency ratio decreases and dew point effectiveness increases as increasing the inlet air relative humidity. The difference between inlet dry bulb temperature and dew point temperature is smaller at high relative humidity, so the dew point effectiveness may increase in a higher relative humidity derived by equation (12). Thus, the dew point effectiveness cannot be considered as a reasonable parameter to characterize the thermal performance of the HMX. When the relative humidity is low, the chemical exergy of inlet air is bigger and the supply air gets more thermal exergy, as the mechanical energy is almost unchanged. Thus, higher exergy efficiency ratio is obtained as the relative humidity is low. Moreover, it can be observed that the exergy efficiency firstly increases towards a maximum value when the inlet air relative humidity is 0.6, and then falls with the increase of inlet air relative humidity. Therefore, both energy and exergy analyses should be considered for the optimization of the process to get the best thermal performance at permissible level of thermodynamic cost.

5. Conclusions

A mathematical model for the counter-flow HMX of dew point IEC has been developed by making a few reasonable hypotheses 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 principle of exergy analysis in the HMX is discussed and the effects of inlet air temperature and relative humidity on the thermal performance and thermodynamics performance of the HMX are performed. The main conclusions can be drawn from the present study as follows:

(1) On account of the second law of thermodynamics, the process of evaporative cooling in the HMX is accompanied by sacrificing the chemical exergy and the thermal exergy partly contained in the secondary air for the sake of increasing the thermal exergy of product air, as the change of mechanical exergy is very small.

(2) The cooling capacity, exergy efficiency, dew point effectiveness and exergy efficiency ratio are simultaneously increasing when the inlet air temperature rises. It means that the better thermal performance and thermodynamic performance of the HMX can be got by increasing the inlet air temperature.

(3) The cooling capacity and exergy efficiency ratio are decreasing with increasing the inlet air relative humidity, while the dew point effectiveness is opposite to that. When the inlet air relative humidity is 0.6, the maximum exergy efficiency is obtained. Furthermore, it is found that the dew point effectiveness cannot be considered as a reasonable parameter to characterize the thermal performance of the HMX with the inlet air relative humidity varying.

For further and better understanding of the heat and mass transfer occurred in the M-Cycle counter-flow configuration, some operational and structural parameters that affect the thermal performance and thermodynamic performance of the counter-flow HMX were quantitatively analyzed in the future work.

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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}]		
D_h	Graetz numbe	a	air
e	specific exergy [Jkg^{-1}]	da	dry air
EER_{ex}	exergy efficiency ratio	db	dry bulb
h	heat transfer coefficient, [Wm^{-2}K^{-1}]	dp	dew point
h_m	mass transfer coefficient [m/s]	ch	chemical
i	enthalpy [J/kg]	f	air suppling to room
l	length [m]	in	inlet
Le	Lewis number	lat	latent
m	mass flow rate [kg/s]	me	mechanical
I	exergy destruction [W]	out	outlet
Nu	Nusselt number	p	primary air
p	theoretical fan power [W]	s	secondary air
P	pressure [Pa]	sat	saturation state
ΔP	pressure loss [Pa]	sen	sensible
Pr	Prandtl number	t	total

q	heat transfer rate [W]	*th*	thermal
Q	cooling capacity [W]	v	water vapor
R	gas constant, [$Jmol^{-1}K^{-1}$]	w	water
Re	Reynolds number	0	reference 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	distance [m]	φ	relative humidity

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