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Effects of Ratio of Dynamic Circulation to Evaporation Rates on Exergy and Cooling Efficiencies an Evaporative Cooler

Behzad Omidi Kashani *

Department of Mechanical Engineering, Faculty of Engineering, University of Birjand, Birjand, Iran.

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ABSTRACT

The expected performance characteristics of a wet media in an evaporative cooler with the specified geometric and material aspects are reducing the dry-bulb temperature and increasing the moisture content of the air outlet. Inlet air conditions are not under the control of the designer or the operator, but the choice of media geometry and fabric, the external factors such as the water circulation rate, and the velocity of air passing through the media could be controlled by the designer. Based on cooling performance for the excelsior of aspen wood pad, the minimum amount of ratio of the static circulation to evaporation rates is about 8 to 12, which has been mentioned in the literature. In this work, for the cellulosic pad by considering the exergy and cooling efficiencies, the optimal ratios of circulation to evaporation rates are presented for different air velocities. It can be seen that under the constant inlet air conditions, by increasing the air velocity as: 0.5, 1.0 and 1.5 m/s, the maximum exergy efficiency values are 0.10, 0.13 and 0.18 respectively and there are some specified values (minimum) for the ratios of water circulation to evaporation rates between 2 and 2.8 for the typical cellulose pad. However, for the same air velocities, maximum cooling efficiencies occur at lower exergy efficiencies, such as 0.03, 0.04, and 0.045, respectively.

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1. Introduction

In hot, dry and desert areas, controlling the temperatures of enclosed spaces with residential and industrial uses, including those related to agricultural and livestock activities has always been considered. Evaporative Cooling Pad (ECP) is one of the most effective and cheapest technologies available to create favorable conditions inside the enclosed spaces due to low energy consumption. The basis of ECP is that the hot and dry outdoor air passes through a wet media and as the water evaporates on the surface of this wet media, the temperature of the passing air decreases [1].

In all researches on ECP performance, the operation of the circulation pump can be divided into two different types:

1- Circulation pump is always working continuously and the performance of the pad is always stable and in the current research, this type of circulation is called static. Most research is of this type.
2- The circulation pump is switched on only in a part of a certain working period and the rest of the period is off, so pad performance does not always remain stable and in this research the pad performance is called dynamic. Very little research has been done on this type.

Many studies and researches have investigated the effect of material and their geometry in wet media (pad), different air velocities passing through the pad and different atmospheric conditions on ECP performance along with static circulation in the literature [2-7].

*Corresponding Author: Behzad Omidi Kashani
Email: B.KASHANI@BIRJAND.AC.IR

Paschold et al. [6] investigated the effect of two different environments such as aspen wood excelsior pad and cellulose (rigid media) pad as well as the effect of air velocity passing through the pad on the physical properties of the flow including characteristic length and Reynolds number and residential time. They reported that for aspen and cellulose pads, the Reynolds number was in the range of 492 to 730 and in the range of 1841 to 2829, respectively [6].

Martinez et al. [5] investigated the effect of wet plasticmaterial compared to other wet materials such as cellulose and the effect of different air velocities on the static pad performance and observed that with increasing air velocity from 0.5 to 2.5 m/s, cooling efficiency decreased from 81 to 70%, but exergy efficiency did not change significantly and remained constant. Matrtinez et al. [5] also investigated the effect of different rates of static circulation on cooling and exergy efficiencies. However, they have not investigated the ratio of circulation rates to evaporation and the effect of this ratio on cooling and exergy efficiencies.

Abohorlu et al. [2] investigated the effect of wet pad of eucalyptus fibers at different air velocities on the cooling efficiency of the pad and reported that by increasing the air velocity from 0.1 to 0.6 m/s, the cooling efficiency of 71% decreased to 49%.

There are other studies to investigate the effect of different rates of static circulation, different air velocities passing through different types of pad and different atmospheric conditions on the performance of ECP in the literature [1, 4, 8-11].

According to standards for aspen wood excelsior pads with a thickness of 0.05 m, the range for the ratio of static circulation to evaporation rates was suggested between 9.5 and 10.5 [1, 11]. So far, for cellulose pads no scientific or practical standard or recommendation for the ratio has been reported.

Franco et al. investigated the effect of different static circulation rates along with the effect of passing air velocities through the cellulose pads on the air pressure drop, mass and heat transfer coefficients and cooling efficiency [8]. They report that by increasing the static circulation rate from 0.1 to 0.25 kg/s-m2, the cooling efficiency does not change much (less than 1%). Also, with increasing air velocity from 0.5 to 4 m/s, due to the reduction of residential time, cooling efficiency decreased from 70 to 60%, and at the same time, the mass and heat transfer coefficients increased from 50 to 300 kg/h and from 20 to 120 W/m2-K (6 times), respectively [8].

Hu et al. [10] investigated the effect of different rates of static circulation on air pressure drop across the pad and cooling efficiency. They report that with increasing circulation rate from 0 to 39 l/min-m2 and for air velocities from 0.5 to 3.5 m/s, the air pressure drop increases by 2 to 5 Pa for PVC media pads [10]. Also, for the same air velocities, by doubling the circulation rate from 19.5 to 39 l/min-m2, the cooling efficiency has increased between 2 to 3% [10].

Karaca et al. investigated the effect of 3 different values of static circulation rate of 2, 4 and 6 l/min-m^2 on the cooling efficiency of cellulose pad [4]. They have reported that the highest and lowest cooling efficiencies (reduction of passing air temperature) are in the circulation circuits of 4 and 6 l/min-m2, respectively, and finally stated that no clear result from the effect of changing the static circulation rate on the cooling performance of the pad [4].

Franco et al. Investigated the effect of static circulation rate on the amount of water retained in 4 different cellulose pads with different geometric characteristics and reported that by increasing the circulation rate from 0.12 to 0.26 l/s-m2, the amount of water retained by the pads (with Identical thicknesses) can be increased from 17 to 44% [9].

According to the mentioned researches, about the effect of static circulation rate, it is observed that no standard reference has been provided to the researchers by the pads manufacturers, so the researchers have chosen the circulation rate according to individual choices and experiences, and sometimes they increased the circulation rate 3 times or even more [4, 8, 10, 12].

On the other hand, very little researches have been done by researchers to investigate the effect of dynamic circulation rate on pad performance [13-15]. Omidi Kashani [15] by using dynamic performance was able to increase the energy efficiency ratio (EER) of a direct type evaporator between 10 to 15%. In a study by Sreeram, he first examined the effect of different static circulation rates of on the performance of a cellulose pad and then turned off the pump and found that until about 5 minutes later, the pad could function without any change in its cooling performance. During this period, due to the shutdown of the circulation pump, the pressure drop of the passing air has decreased by 12 Pa (which means an increase in the air passing rate, and of course, an increase in the cooling load was happened) [14].

In other words, it can be predicted that by increasing the cooling load on the one hand and reducing the electrical energy consumption (due to the shutdown of the circulation pump) on the other hand, the energy efficiency ratio (EER) of the pad with dynamic performance can be increased than its static performance [15].

In a study conducted by Rong et al., They studied the performance of cellulose pads with dynamic circulation rate and calculated the cooling efficiency the heat and mass transfer coefficients [13]. Rong et al. are one of the few researchers to investigate the effects of dynamic circulation rate variations and different air velocities on cooling performance in a cellulosic pad, to regulate the dry-bulb temperature and relative humidity of the outlet air [13]. They varied the pump-on time (p.o.t.) duration between 3 and 120 seconds in different time control cycles (which include the durations of pump-on and pump-off). The control time cycles (c.t.c.) are chosen as 3, 4, and 5 minutes [13].

They examined the effect of 21 different dynamic circulation rates from 5.6 to 300 l/h at 3 different air velocities from 0.5 to 1.5 m/s on pad performance (cooling efficiency and mass transfer coefficients). They observed that with increasing dynamic circulation rate, the cooling efficiency of the pad for all passing air tracks increases from 45 to 60% and with the increase of the passing air speed, the cooling efficiency decreases by 20 to 25%.

According to these researches, it can be seen that the simultaneous effects of important parameters such as dynamic circulation rate and air velocity on energy and exergy efficiencies and evaporation rate have not been studied at all. The circulation rates selected by previous researchers were based on personal taste and experience and is not based on any scientific and standard principles based on the geometry and material of the pad.

The purposes of the present study are: a) to determine the maximum amount of evaporation rate (nominal evaporation capacity) for a type of cellulose pad and b) to determine the allowable range of circulation rate (or ratio of circulation rate to evaporation rate) according to the conditions of maximum cooling and exergy efficiencies of the pad. In the present work, for a typical cellulosic pad, the evaporation rate, the cooling and exergy efficiencies are investigated for different dynamic water circulation rates and different air velocities based on result of Rong et al. [13].

2. Materials and methodologies

2.1. Extracted experimental data

Due to limited laboratory and experimental facilities, the calculations in the present work are based on the extraction of experimental data from Rong et al. research [13]. As mentioned in the introduction, for a pad with dynamic performance, a suitable range of circulation rate (or the ratio of circulation to evaporation rates) for different air velocities has not been proposed so far. The selected ECP made by impregnated and corrugated cellulose paper sheets with various flute angles available from Munters AB, Kista, Sweden based on Rong et al. research [13]. The primary experimental results of Rong et al. include:

- Instantaneous values of dry bulb temperature and relative humidity of the inlet and outlet air of the wet pad for each air velocity,
- Instantaneous cooling efficiency,
- Instantaneous mass transfer coefficient in each of the 63 different operating conditions (according to Table 1, there are 21 cases for each air velocity, then for three different passing air velocities, there are total 21x3 = 63 cases).

Then the data was extracted from the resulting curves of Rong et al [13] by Plot Digitizer software [16]l}. The different amounts of dynamic water circulation rates are found by different amounts of control time cycle (c.t.c.) and pump on-time (p.o.t.). 3 different values of c.t.c. are 3, 4 and 5 minutes and 7 different p.o.t values as 3, 5, 10, 20, 30, 60 and 120 seconds are chosen by Rong et al [13]. The static water circulation rate is 7.5 l/min. For example, if the amounts of c.t.c. and p.o.t. values are chosen as 4 minutes and 3 seconds, respectively, the dynamic circulation rate is 7.5×3/4/60 = 0.125 l/min, which is about 1:80 smaller than static circulation rate. Certainly, there was the possibility of hot spots occurrence on the pad for this amount of circulation rate as static kind, but not for dynamic one. As mentioned before, in the present work, the experimental values of the outlet air dry-bulb temperature of the cellulose pad used from Rong's results [13] values for c.t.c. and seven different p.o.t. values, so totally, there are 21 different dynamic water circulation rates, according to Table 1 for each air velocity.

By ascend arranging the different states in Table 1 according to the values of the dynamic water circulation rate as shown in Table 2, it can be seen that the dynamic water circulation rate could be varied from 4.5 l/h to 300 l/h and also the ratio of dynamic to static circulation rates could vary from 0.01 to 0.667. It can be seen that the reduction of the circulation rate to about 0.01 compared to the static circulation rate is only possible with the dynamic method.

During this periodic dynamic performance of the pad, the dry-bulb outlet temperature and the cooling efficiency are instantly measured in different air velocities, and a relationship between the dynamic circulation rate, and air passing rate, and cooling efficiency was provided for the typical pad [13. For each amount of air velocity and all different dynamics circulation rates (21 cases), the conditions of inlet air (including dry-bulb air temperature and relative humidity) were kept constant at specified amounts, as mentioned in Table 3.

So, in the present work, only the effects of external factors (such as the circulation rate and the air velocity that can be controlled by the operator of the cooler) on pad performance (such as evaporative rate, thermal performance, energy and exergy efficiencies) have been investigated.

Here, it is assumed that the cooling evaporation process is carried out as the adiabatic process. Therefore, the wet-bulb temperature of the air remains constant through passing the pad.

2.2. Mathematical Equations

In the present work, the cooling and exergy efficiencies and the exergy values of inlet and outlet air and water through the pad are calculated.

Table 1. Dynamic and static circulation rates and their ratios for different amounts of c.t.c. and p.o.t [13]

case	c.t.c. (min)	p.o.t (sec)	dynamic water circulation rate (lit/h)	static water circulation rate (lit/h)	the ratio of dynamic to static water circulation rates
1	3	3	7.5	450	0.017
2	3	5	12.5	450	0.028
3	3	10	25	450	0.056
4	3	20	50	450	0.111
5	3	30	75	450	0.167
6	3	60	150	450	0.333
7	3	120	300	450	0.667
8	4	3	5.625	450	0.0125
9	4	5	9.375	450	0.021
10	4	10	18.75	450	0.042
11	4	20	37.5	450	0.083
12	4	30	56.25	450	0.125
13	4	60	112.5	450	0.25
14	4	120	225	450	0.5
15	5	3	4.5	450	0.01
16	5	5	7.5	450	0.017
17	5	10	15	450	0.033
18	5	20	30	450	0.067
19	5	30	45	450	0.1
20	5	60	90	450	0.2
21	5	120	180	450	0.4

The evaporative cooling process is modeled under insulation conditions, so the wet-bulb air temperature is assumed to be constant during the passage of air. The temperature of the inlet water is assumed to be the same as the wet-bulb temperature of the entering air. The following relation can find cooling efficiency ($\eta_{cooling}$):

$$\eta_{cooling} = \frac{(T_{in} - T_{out})}{(T_{in} - T_{in,wb})} \quad (1)$$

where T_{in}, T_{out}, and $T_{in,wb}$ are the inlet air dry-bulb, the outlet air dry-bulb, and the inlet air wet-bulb temperatures, °C, respectively. In the present work, T_{wb} represents the wet-bulb temperature, (°C), and is calculated based on the values of dry-bulb temperature, T (°C), and relative humidity, φ (%), by following equation [17].

$$\begin{aligned} T_{wb} = T\,.\,tg^{-1}\,[&0.151977(\varphi \times 100 \\ &+ 8.31659)^{0.5}] \\ &+ tg^{-1}(T + \varphi \times 100) \\ &- tg^{-1}(\varphi \times 100 - 1.676331) \\ &+ 0.00391838(\varphi \times 100)^{1.5} \\ &\times tg^{-1}(0.023101 \times \varphi \times 100) \\ &- 4.686035 \end{aligned} \quad (2)$$

Table 2. Dynamic water circulation rates for the 21 cases in ascending order [13]

case	dynamic water circulation rate (lit/h)	the ratio of dynamic to static circulation rates of water
15	4.5	0.01
8	5.625	0.0125
1	7.5	0.017
16	7.5	0.017
9	9.375	0.021
2	12.5	0.028
17	15	0.033
10	18.75	0.042
3	25	0.056
18	30	0.067
11	37.5	0.083
19	45	0.1
4	50	0.111
12	56.25	0.125
5	75	0.167
20	90	0.2
13	112.5	0.25
6	150	0.333
21	180	0.4
14	225	0.5
7	300	0.667

Table 3. Conditions of inlet air to the pad at different inlet air velocities [13]

Air velocity (m/sec)	Inlet air dry-bulb temperature (°C) (average)	Inlet air relative humidity - RH (%) (average)
0.5	45.8 – 46.5 (46.15)	3.6 – 4.0 (3.8)
1.0	44.5 – 45.0 (44.75)	4.1 – 4.9 (4.5)
1.5	40 – 45.3 (42.65)	5.4 – 8.3 (6.85)

The values of dry-bulb temperature and relative humidity for inlet air are known, so the inlet air wet-bulb temperature is explicitly obtained from equation (2). But for the outlet air, based on the adiabatic assumption of the process, the outlet air wet-bulb temperature is assumed to be equal to the inlet air wet-bulb temperature, and it is known (as mentioned before). In each case, by applying the known values of wet-bulb temperature, T_{wb}, and outlet air dry-bulb temperature, T, from the experimental work of Rong et al. [13] (Rong et al., 2017), the relative humidity of the outlet air from equation (2) is implicitly found in terms of T_{wb} and T. The amount of evaporative water rate can be found from the following relation:

$$\dot{m}_v = \dot{m}_{a2} \,.\, \omega_2 - \dot{m}_{a1} \,.\, \omega_1 \tag{3}$$

where $\dot{m}_{a1}$ and $\dot{m}_{a2}$ are the inlet and outlet dry air mass rates, respectively, and ω_1 and ω_2 are the kg of water vapor/kg dry air, respectively. $\dot{m}_a$ and ω are also found from the following relation:

$$\dot{m}_a = \frac{(P - \varphi \,.\, P_s) \,.\, V}{287 \,.\, T} \tag{4}$$

$$\omega = 0.622 \, \frac{\varphi \,.\, P_s}{(P - \varphi \,.\, P_s)} \tag{5}$$

where P, φ, P_s, V, and T are atmospheric pressure equal to 101325 Pa, relative humidity, water vapor saturation pressure (Pa), the volume rate of passing air, and air dry-bulb temperature in sections 1 and 2, respectively for inlet and outlet sections. The water vapor saturation pressure, P_s, is expressed in terms of Pascal (Pa) from the following relation [18]:

$$P_s = 610.8 \,.\, \exp\left(\frac{T}{T + 238.2} \times 17.2694\right) \tag{6}$$

T is the dry-bulb air temperature of °C, V is the volume rate of air that is equal to the product of the velocity of air passing through the front surface area (air passage), and the front surface area (0.6×1.8 = 1.08 m2).

2.2.1. Exergy Analysis

According to the Figure 1, the inlet air has index 1 and the outlet air has index 2, and the inlet circulation water is entered into the pad has index 3 and part of this water is evaporated ($\dot{m}_v$) and the surplus of this water is just exited at the bottom of the pad with index 4 and this water discharged into the water collection container. The water circulation rate is always higher than the rate of evaporation water. The water circulation rate can be calculated as follows:

$$\dot{m}_{w,3} = dynamic\ circulation\ rate = \dot{m}_v + \dot{m}_{w,A} \tag{7}$$

Exergy is the ability to convert energy into work. Exergy can be transmitted in three ways: work, heat transfer, and mass transfer through the control surface. All processes are irreversible essentially, and some exergy is destroyed during these processes.

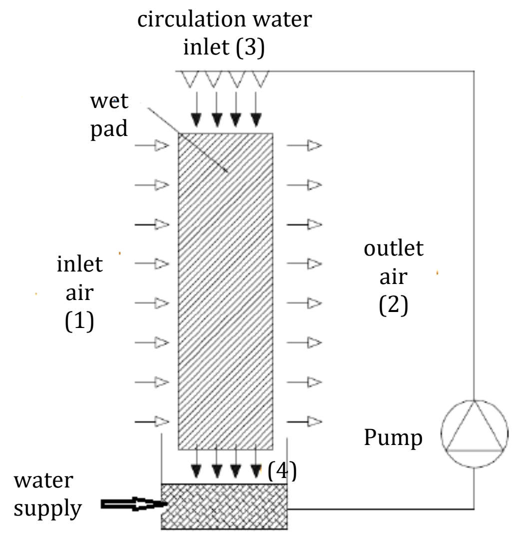

Figure 1. Overview of the control volume of a wet pad in of a direct evaporative water cooler

The exergy balance for a wet pad as a control volume, in steady-state conditions where there is no work exchange with its environment, could be written as [19-20]:

$$\sum_{in} \dot{E}x_Q + \sum_{in} \dot{m} \,.\, \psi - \sum_{out} \dot{E}x_Q - \sum_{out} \dot{m} \,.\, \psi - \dot{E}x_{dest} = 0 \tag{8}$$

where $\dot{E}x_Q$ and $\dot{E}x_{dest}$ are the amount of exergy exchanged due to heat transfer and the amount of exergy destruction (is always positive) during this process, respectively. Also, ψ is the specific amount of fluid exergy at each inlet and outlet flows.

2.2.1.1 Exergy of inlet and outlet air through the pad

The exergy values of air are expressed as the equation (9) as follows [20]:

$$\psi_a = \psi_{phy} + \psi_{chem} \tag{9}$$

The amount of specific physical exergy ($\boldsymbol{\psi_{phy}}$) according to equation (9-1) can be divided into two parts: thermal and mechanical, as follows:

$$\psi_{phy} = \psi_{phys-thermal} + \psi_{phys-mechanical} \tag{9-1}$$

The contributions of physical-thermal and physical-mechanical exergies are calculated by the equations (9-2) and (9-3), respectively as below:

$$\psi_{phy-thermal} = (Cp_a + \omega\, Cp_v)T_0 \left(\frac{T}{T_0} - 1 - \ln\left(\frac{T}{T_0}\right)\right) \tag{9-2}$$

$$\psi_{phys-mechanical} = (1 + 1.608\omega)R_a T_0 \,.\, \ln\left(\frac{P}{P_0}\right) \tag{9-3}$$

Since the wet pad is at ambient pressure and except for the slight pressure drop during air passage, it can always be considered at ambient pressure (P = P0), so the physical-mechanical exergy contribution of the air is always zero.

The chemical exergy value of air is also given by equation (9-4) as follows:

$$\boldsymbol{\psi_{chem}} = \boldsymbol{R_a T_0}\left[(\mathbf{1} + \mathbf{1.608}\boldsymbol{\omega} \,.\, \boldsymbol{ln}\left(\frac{\mathbf{1} + \mathbf{1.608}\boldsymbol{\omega_0}}{\mathbf{1} + \mathbf{1.608}\boldsymbol{\omega}}\right) + \mathbf{1.608}\boldsymbol{\omega} \,.\, \boldsymbol{ln}\left(\frac{\boldsymbol{\omega}}{\boldsymbol{\omega_0}}\right)\right] \tag{9-4}$$

in the above equations, Cp_a, Cp_v , ω, ω_0 , T_0 and R_a are the heat capacity of the air, the heat capacity of the water vapor, the ratio of humidity, the ratio of humidity at dead state (which the system is in equilibrium with temperature, pressure, and chemical equilibrium), dry-bulb air temperature, base temperature (the same as the ambient dry-bulb temperature or inlet air dry-bulb temperature to the pad or at dead state), K, and the gas constant for air, respectively.

Dead conditions for chemical exergy are T0 as atmosphere dry-bulb temperature (inlet air dry-bulb temperature), P0 atmospheric pressure (constant and equal to 101325 Pa), and $\boldsymbol{\omega_0}$ as the ratio of humidity at inlet air wet-bulb temperature.

The contribution of the physical-thermal exergy of inlet air to the pad according to equation (9-2) depends on air dry-bulb temperature and air humidity ratio. As the dry-bulb air temperature and the dry-bulb temperature of the base point are the same, the above term is always zero for inlet air. By eliminating the physical-mechanical exergy term of the air, the only exergy component for the inlet air is the chemical exergy contribution which is found from equation (9-4).

2.2.1.2. Specific exergy for inflow and outflow of water through the pad

Inlet and outlet water temperatures are equal to the inlet air wet-bulb temperature to the pad, the water flow exergy is calculated according to the following simplified equation [21];

$$\boldsymbol{\psi_\omega} = -\boldsymbol{R_v T_0}\ \mathbf{ln}\ (\boldsymbol{\phi_0}) \tag{10}$$

here, R_v is gas constant for vapor, and ϕ_0 is the ratio of the water vapor pressure of the unsaturated atmospheric air to the saturated water vapor pressure at the dead state temperature. According to Figure 1 and the presence of two inlet mass flows (sections 1 and 3) and two output mass flows (sections 2 and 4), exergy efficiency is found by two different ways as below:

(a) The general definition of exergy efficiency can be as follow:

$$\boldsymbol{\eta_{II}} = \frac{\sum_{out}\dot{\boldsymbol{m}}\,.\,\boldsymbol{\psi}}{\sum_{in}\dot{\boldsymbol{m}}\,.\,\boldsymbol{\psi}} = \frac{\dot{\boldsymbol{m}}_2\,\boldsymbol{\psi}_2 + \dot{\boldsymbol{m}}_4\,.\,\boldsymbol{\psi}_4}{\dot{\boldsymbol{m}}_1.\boldsymbol{\psi}_1 + \dot{\boldsymbol{m}}_3\,.\boldsymbol{\psi}_3} = \frac{\dot{\boldsymbol{m}}_2\,.\boldsymbol{\psi}_2 + \dot{\boldsymbol{m}}_{add}\,.\boldsymbol{\psi}_4}{\dot{\boldsymbol{m}}_1\,.\boldsymbol{\psi}_1 + (\dot{\boldsymbol{m}}_v\,.\boldsymbol{\psi}_3 + \dot{\boldsymbol{m}}_{add}\,.\boldsymbol{\psi}_3)} \tag{11}$$

And since the specific exergies are equal in sections 3 and 4, the above relation can be written as follow:

$$\eta_{II} = \frac{\dot{m}_2\psi_2 + \dot{m}_{add}\,.\psi_3}{\dot{m}_1.\psi_1 + (\dot{m}_v.\psi_3 + \dot{m}_{add}\,.\psi_3)} \tag{12}$$

(b) According to the target term in equation (12), which is the output exergy term in section 2, the target exergy efficiency can be defined by the following equation (13):

$$\boldsymbol{\eta_{II,*}} = \frac{\sum_{out,*}\dot{\boldsymbol{m}}.\boldsymbol{\psi}}{\sum_{in}\dot{\boldsymbol{m}}\,.\boldsymbol{\psi}} = \frac{\dot{\boldsymbol{m}}_2\boldsymbol{\psi}_2}{\dot{\boldsymbol{m}}_1\boldsymbol{\psi}_1 + (\dot{\boldsymbol{m}}_v.\boldsymbol{\psi}_3 + \dot{\boldsymbol{m}}_{add}\,.\boldsymbol{\psi}_3)} \tag{13}$$

where the sign (*) relates to the target exergy flow, by comparing the equations (12) and (13), it was seen, that there is always a relationship between these two exergy efficiencies as below:

$$\boldsymbol{\eta_{II,*}} < \boldsymbol{\eta_{II}} \tag{14}$$

Term $\dot{E}x_{dest}$ can be rewritten from the exergy balance, equation (8) as follows:

$$\dot{E}x_{dest} = \sum_{in}\dot{E}x_Q + \sum_{in}\dot{m}\,.\psi - \sum_{out}\dot{E}x_Q - \sum_{out}\dot{m}\,.\psi = \sum_{in}\dot{m}\,.\psi - \sum_{out}\dot{m}\,.\psi \tag{15}$$

The Overall exergy efficiency can be defined as equation (16):

$$\eta_{II} = 1 - \frac{\dot{E}x_{dest}}{\sum_{in}\dot{m}\,.\psi} \tag{16}$$

there is no similar result, such as the last equation for $\eta_{II,*}$. Here, in terms of the purpose of evaporative cooling, which is to reduce the dry-bulb temperature and to increase the humidity of the outlet air through the pad (section 2), equation (13) is used to define the targeted exergy efficiency (from now as the preferred exergy efficiency).

A review of some previous works has shown that for calculating the exergy efficiency, the exergy of water which introduced in section 3, Figure 1, referred to the exergy of water consumed (evaporated) and the exergy associated with excess water flows over evaporation in that section was not considered, so there was not any outlet water flow at section 4 [21-22]. But in the results of some other researchers, the effect of exergy related to surplus water on evaporation in sections 3 and 4 has been considered in calculating the exergy efficiency [23].

When examining the effect of the circulation rate as a variable parameter on the pad exergy efficiency, it is necessary to consider the exergy of the surplus water flow rate (Section 4) in both numerator and denominator of the exergy efficiency fraction according to equation (13). In the present work, however, in terms of more considerable attention to outlet air conditions in the targeted exergy efficiency (equation (13), the exergy associated with excess water flow to the evaporated is considered only at the denominator of this equation (as part of exergy in section 3).

2.3. Series of calculation steps

All necessary thermodynamic variables, including absolute humidity at the inlet and outlet, the rate of evaporated mass, the mass of excess water leaving the pad, enthalpy, entropy and heat capacity of liquid water and steam were calculated by a computer program which was written in engineering equation solver (EES) software and series of calculation steps are summarized as below:

A. With the experimental values of dry-bulb temperature and relative humidity of the inlet air and applying equation (2), the wet-bulb temperature of inlet air to the pad is found and then based on the assumption of an adiabatic process for the control volume around the pad according to Figure 1, the wet-bulb temperature of outlet air is the same as the wet-bulb temperature of inlet air to the pad. Then, with the known variables of wet-bulb temperature and dry-bulb temperature of the output and inlet air of the pad (sections 1 and 2 at Figure 1), the absolute humidity of the inlet and outlet air of the pad are found by the EES code.
B. With the differential amount of the absolute humidity of the outlet and inlet air and mass rate of passing air (based on the velocity of passing air through the pad), the evaporation rate (rate of water supply rate) is found.
C. By subtracting the evaporation rate from the dynamic circulation rate, the excess water output rate from the bottom of the pad is found (section 4 of Figure 1).
D. The inlet and outlet water temperatures at sections 3 and 4 of Figure 1 are assumed to be equal. Based on the values of air and water inlet and outlet rates to the control volume, both of the cooling and exergy efficiencies are calculated for each case by the EES code [24]. The process of calculating the rate of evaporation rate and energy and exergy efficiencies is given in the flowchart in Figure 2.

3. Results and discussion

The least expensive and safest tool to evaluate the performance of the pad is to calculate the efficiency of the first law (cooling efficiency) and the efficiency of the second law of thermodynamics (exergy efficiency) based on the known properties of parameters in sections 1 to 4 of Figure 1.

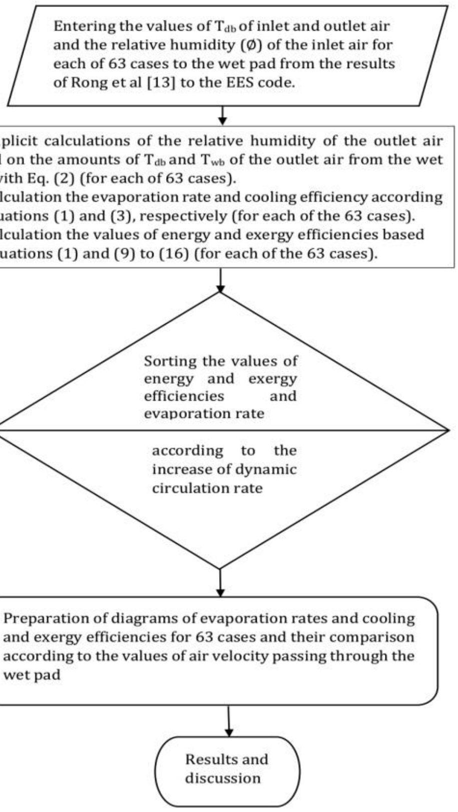

Figure 2. Flowchart of procedure of calculations performed by the EES code in the current study

Here, the pad is considered as a control volume or a black box that does not need to be examined in detail the interactions of its internal phenomena (related to the mass and heat transfer processes, capillary, and water absorption capacity of the pad and the effect of air velocity on the transmission surface). As previously mentioned in the introduction, for the pad with dynamic condition, the exergy efficiency has not been investigated so far.

For calculating the cooling efficiency, only the effect of sensible heat transfer to the passing air is considered. But in exergy efficiency, in the chemical term of exergy, the degree of chemical effectiveness (increasing the humidity of the passing air - mass transfer) and in the thermo-mechanical term, the sensible heat transfer (due to the change in dry air temperature-heat transfer) in the exergy of the passing air are considered, simultaneously. So, the simultaneous occurrence of the maximum of the energy and exergy efficiencies of pad performance is impossible. Here, the ranges of the new parameter (ratio of circulation to evaporation rates) have been investigated for the occurrence of maximum values of cooling and exergy efficiencies (first and second laws of thermodynamics) according to the air velocity passing through the pad.

The effect of dynamic water circulation rate on cooling efficiency and evaporated water rate for air velocity of 0.5 m/sec are shown in Figure 3.

In Figure 3, the similarity of the behavior of these two curves is well seen. It is also understood that by increasing the dynamic water circulation rate to a certain limit, about 50 kg/h, the cooling efficiency and evaporation rate increase dramatically, and from there on, the initial slopes of the curves decrease simultaneously and visibly. So, with increasing circulation rates up to 240 kg/h (about five times the limit value), only about 14 and 16% increment in average evaporation rate and cooling efficiency were happened, respectively. Point P and B, represent the circulation rates for occurrence states of the maximum exergy and cooling efficiencies, respectively. The circulation range C from P to B is suggested as an acceptable range for the typical pad with 0.5 m/s air velocity.

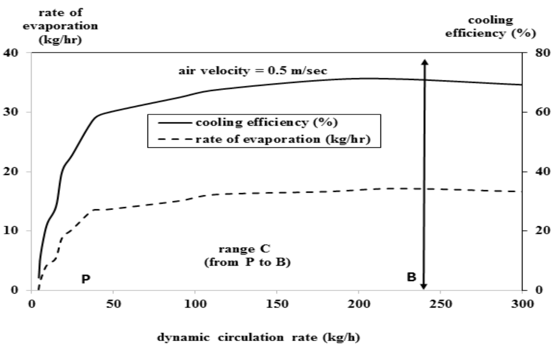

Figure 3. Evaporation rate and cooling efficiency in terms of dynamic circulation rate for 0.5 m/sec air velocity

The curves of cooling efficiency and evaporated water rate in terms of the dynamic circulation rate for all air velocities as 0.5, 1.0, and 1.5 m/sec, are given in Figures 4 and 5, respectively. In Figure 4, it is seen: first (same as Figure 3) there are an initial slope of the curves up to a specific circulation rate (point P, about 37.5 to 45 kg/h), and second cooling efficiency was reduced by increasing the air velocity which was due to the reduction of residental time of the passing air in near the surface of the wet pad, which is consistent with the literature.

And third, up to point B (range 240 to 300 kg/h, respect to all air velocities) the cooling efficiency (or evaporation rate) increased about 14 to 16% than point P. The circulation rate at point B is the nominal circulation rate (maximum evaporation rate happens in this circulation rate).

Figure 5 shows the effect of the dynamic circulation rate on the evaporation rate for different velocities of air passing through the pad. The same behavior as in Figures 3. and 4 is again observed for the curves of water evaporation rates at a specified circulation rate of (point P, with 37.5 to 45 kg/h circulation rate), but with increasing the air velocity the evaporation rate increases as a result of higher mass rate of passing dry air and the evaporation intensification is happened due to the increased mass transfer coefficient due to the increasing Reynolds number and this behavior is expected as in the relevant literature.

Based on Figure 5, the nominal capacity (maximum evaporation rate) which happened at maximum cooling efficiency, is 35.48 kg/h at 300 kg/h circulation rate at 1.5 m/s air velocity for a typical pad. So, the nominal ratio of circulation to evaporation rates is 8.45. The evaporation rate between the occurrence position of maximum of exergy efficiency (point P as seen in Figure 8) and the nominal capacity of the pad (point B) changes only about 14% to 20%.

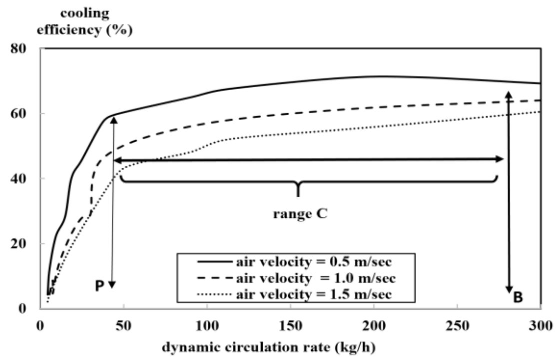

Figure 4. Cooling efficiency in terms of dynamic water circulation rate for different air velocities through the pad.

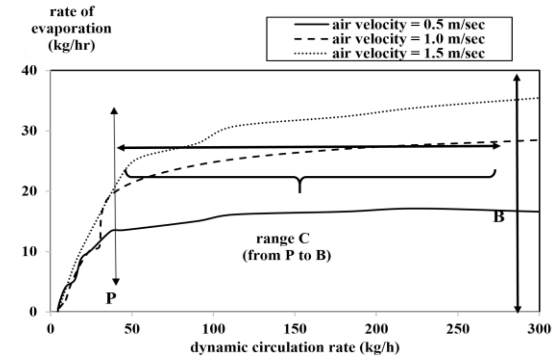

Figure 5. Evaporated water rate in terms of dynamic water circulation rate at different air velocities through the pad.

It can be seen from Table 3, dry temperature and relative humidity of the inlet air to the wet pad are kept almost the same and constant for different air velocities.

Under these conditions, the changes in dry air temperature and relative humidity of the outlet air from the wet pad in terms of dynamic circulation rate are shown in Figures 6 and 7, respectively. In these figures, it can be seen that by increasing the speed of the air passing through the wet pad, the residential time of passing air decreases near the pad and the outlet air does not have enough opportunity to reduce its temperature and gain more moisture. Therefore, with increasing the velocity of the passing air, the dry temperature of the outlet air increases and the relative humidity of the exhaust air also decreases.

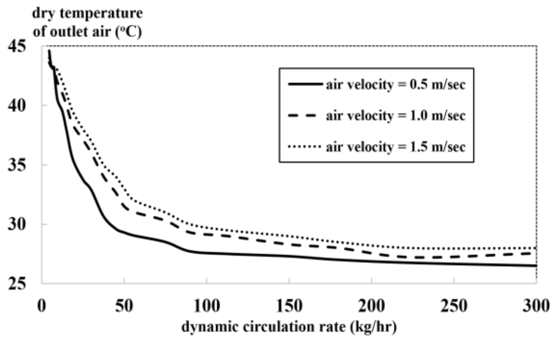

Figure 6. Outlet air dry temperature at different dynamic circulation rates

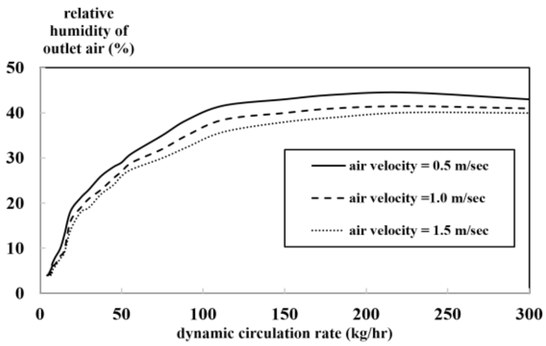

Figure 7. Relative humidity of outlet air at different dynamic circulation rates

Figure 8 shows the targeted exergy efficiency in terms of the dynamic water circulating rate for different velocities of air through the pad. As mentioned earlier, in Figures 3 to 5, the maximum of exergy efficiency occurred at the point P (with 37.6 to 45 kg/h circulation rate). For larger amounts of dynamic water circulation rate (such as point B, the circulation rate for the maximum cooling efficiency), the exergy efficiency drops. Range C is formed from the least circulation rate at point P to the highest circulation rate at point B. The exergy efficiency drop can be due to factors such as:

(i). Decrease of the practical evaporation level due to the thickening of the layer of water coated on the wet pad surfaces or

(ii). Different interact of heat and mass transfer phenomena from the pad to the passing air (depending on the velocity of the passage of air), as it was seen before in Figure 8, the exergy efficiency increases as the air velocity increases.

The exciting thing about the behavior of the typical pad (depending on its geometry and material) is that for this range of air velocities between 0.5 m/sec and 1.5 m/sec, maximum exergy efficiency always occurs at the specified circulation rate (37.5 to 45 kg/h). Similar investigations on the pad performance with different geometry, material, and air velocities could be useful for the industrial designers and customers of the wet pad.

In Figure 9, the curves of the ratio of circulation to evaporation rates are shown in terms of the dynamic circulation rate for different air velocities. It is seen that the minimum amount of the ratio is about 2.8, 2, and 2 for different air velocities as 0.5, 1.0, and 1.5 m/sec, respectively, occurred at the same amount of circulation rate (37.5 to 45 kg/h where the maximum exergy efficiency has happened). In other words, this specific behavior of the pad indicates that at this amount of circulation rate, an optimal and relevance interacts happened between the evaporation and the cooling phenomena of the passing air for all different air velocities. At this special condition (position P), the maximum of exergy efficiency is happened.

For this pad, a minimum circulation rate of 50 kg/h is suggested and it is recommended that the circulation rate can be higher than this specified value (for working conditions). Figures 5. and 9. show that by a 6-fold increment of the circulation rate from point **P** to point **B** (50 kg/h to 300 kg/h), the ratio of the circulation to the evaporation rates increased simultaneously, but the evaporated rate only increased approximately 14% to 42% (from point **P** to **B**) for all passing air velocities. The rectangle abcd is the optimal domain for working condition of the pad. There are two horizontal lines at **P** and **B**, which represent the ratio of circulation to evaporation rates for occurrence the maximum exergy and energy efficiencies, respectively. And also, there are two vertical lines at **P'** and **B'**, which are representative the rate of circulation for the maximum exergy and cooling efficiencies, respectively.

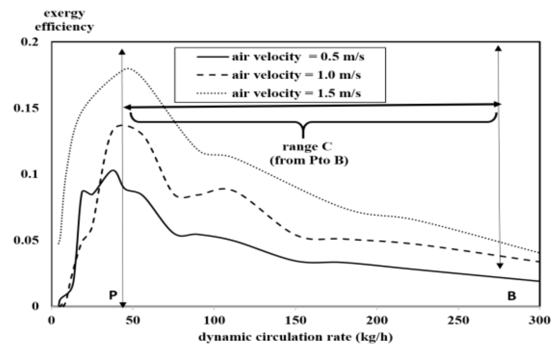

Figure 8. Exergy efficiency (Targeted) in terms of dynamic water circulation rate for different air velocities through the pad.

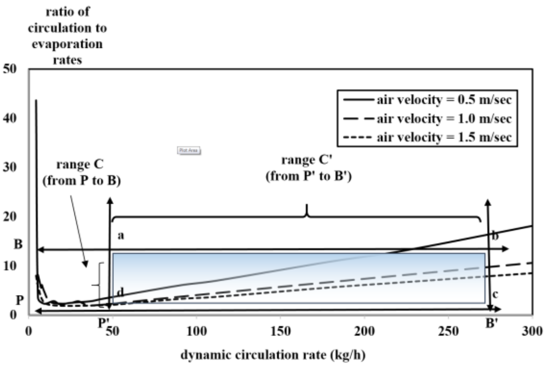

Figure 9. The ratio of the dynamic water circulation to water evaporated rates for different amounts of water circulation rate

In Figure 10, the cooling efficiency is presented in terms of the ratio of circulation to evaporation rates for different air velocities. The ranges **P** and **B**, represent the relevant ratios of circulation to evaporation rates for occurrence of maximum exergy efficiency (from 1.93 to 2.80) and cooling efficiency (from 8.50 to 13.15) for all air velocities, respectively.

Therefore, for the optimal operation of a typical pad (with specific material and geometric characteristics) in the intermediate conditions of maximum cooling and exergy efficiencies, a certain range (**C**) of the new parameter is recommended.

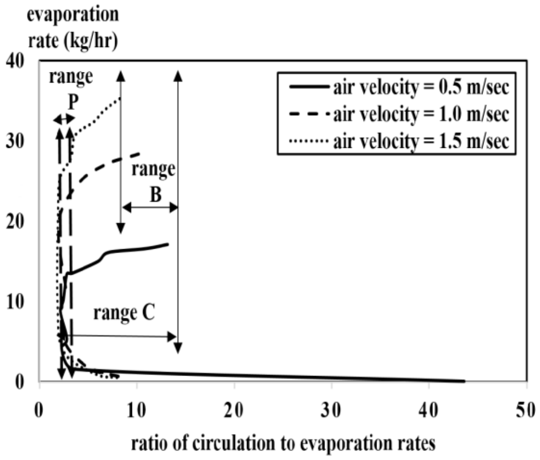

Figure 10. Cooling efficiency in terms of the ratio of the circulation to the evaporated rates

This range (**C**) is obtained by jointing the two ranges **P** and **B** (related to maximum cooling and exergy efficiencies, respectively). The range **C** is formed from 1.93 to 13.15 (from the lowest amount of range **P** to highest amount of the range **B)**. The pad performance conditions are somehow between the two conditions associated with the maximum exergy and cooling efficiencies. The nominal capacity (maximum evaporation rate) of the typical pad can be determined by the pad manufacturers in the most challenging working conditions (according to the climatic conditions and working days during the year and the passing air velocities).

The curves in Figure 11 have a non-functional behavior same as Figure 10, but similar to Figure 5, the evaporation rate increases with higher air velocity. Also the maximum evaporation rate occurs at maximum of the cooling efficiency conditions, not at the maximum exergy efficiency condition. Range **P** and **B**, again represent the conditions for occurrence the maximum of exergy and cooling efficiencies, respectively. And the range **C** is formed by jointing of the two ranges **P** and **B**.

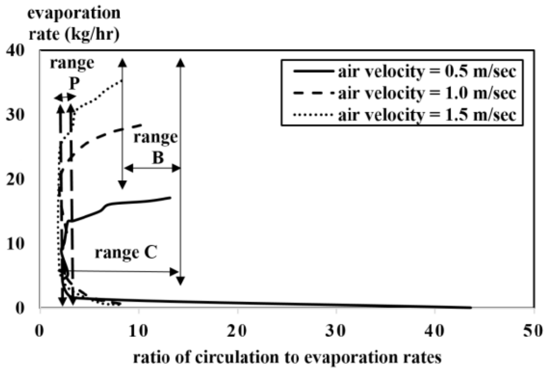

Figure 11. Water evaporation rate in terms of the ratio of circulation to evaporation water rates

Figure 12 shows the variation of exergy efficiency in terms of the ratio of the circulation rate to the evaporation rate for different velocities of passage air, as well as the position and values of the maximum exergy efficiency. It can be seen that the maximum exergy efficiency in the ratio of circulation rate to evaporation rate was in the range of 1.93 to 2.80. Also with increasing the air velocity, the maximum exergy efficiency value increases as 0.10, 0.13 and 0.18 for 0.5, 1.0, and 1.5 m/sec, respectively. It seen, there are a unique non-functional behavior of the curves, so for reaching the minimum amount of the ratio of circulation to evaporation rates, by decreasing the ratio through both the lower and upper paths, the maximum of exergy efficiency is achieved.

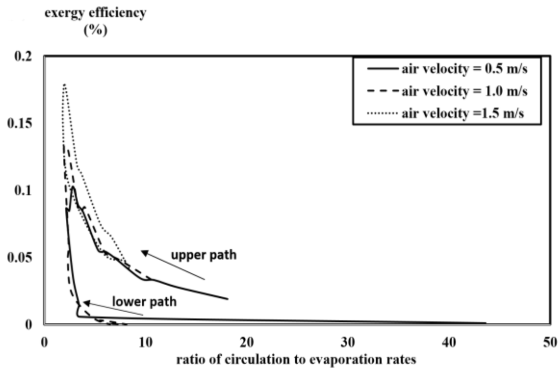

Figure 12. Exergy efficiency in terms of the ratio of circulation to evaporation rates for different air velocities

Finally, in Figure 13, the exergy efficiency is plotted in terms of cooling efficiency, and it seen that the maximum of exergy efficiency occurs when the cooling efficiency is not at its maximum value, and vice versa. By increasing air velocities, the maximum exergy and cooling efficiencies increase and decrease, respectively. The maximum cooling efficiency occurs where exergy efficiency occurs at far below their maximum values. The maximum of exergy efficiency is 0.10, 0.13, and 0.18 in these conditions, where the cooling efficiencies are 57.3%, 46.3%, and 40.8% for the air velocities of 0.5, 1.0, and 1.5 m/s, respectively.

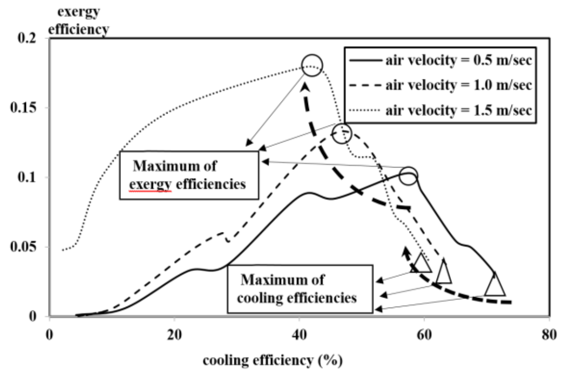

Figure 13. Exergy efficiency in terms of cooling efficiency for different velocities of air passing through the pad.

The maximum feasible of cooling efficiencies were 71.2%, 65.2%, and 60.6% for air velocities of 0.5, 1.0 and, 1.5 m/s, respectively, where the exergy efficiencies as 0.03, 0.034, and 0.04. Increasing the air velocity has three different effects on the pad performance: First it reduces the residential time of the air (reducing cooling efficiency and increasing evaporation rate), second the air velocity, the geometry and the kind of pad material, and circulation rate have complicate effects on effective contact surface area of the pad (for heat and mass transfer phenomena) and third, by changing the air velocity (and changing the Reynolds number) both the mass transfer coefficients (evaporation) and the heat transfer coefficient (cooling phenomenon) would be changed. Exergy efficiency can also be interpreted as affecting the air passing through the pad by lowering its temperature and absorbing water vapor. For cellulose and aspen wood-based pads, no minimum amount of static or dynamic water circulation rate based on the exergy approach has been recommended yet.

4. Conclusion

1. Comparing Figures 3 to 5 with Figure 8 shows that at point P, which is at the end of the steep slope of the cooling efficiency curves, it is exactly where the exergy efficiency of pad performance is maximized.
2. According to Figures 10 and 12, the simultaneous effect of the ratio of circulation to evaporation rates and the velocity of the passing air on the energy and exergy efficiencies of the pad were observed, respectively. Theses curves have not been achieved in previous researches on the ECP. Similar curves can be obtained for any other type of pad depending on its material and geometry, which can be applicable for pad designer and end users
3. For the typical pad was investigated here, the nominal capacity (maximum evaporation rate), the nominal ratio of dynamic circulation to evaporation rates, the recommended range of the dynamic circulation rate, and the range of exergy and cooling efficiencies are 37.5 kg/h, 13.15 and from 37.5 to 300 kg/h, from 0.027 to 0.18, and 41% to 71%, respectively.
4. The velocity of air, effects on residential time of passing air through the pad, and also on the interaction between the mass transfer (evaporation) and heat transfer phenomena. The change of the dynamic circulation rate also impacts on the creation of an appropriate surface for the evaporation phenomenon, it is clear that in substantial amounts of the circulation rate, the flood phenomenon occurs, such as residential time and surface area for evaporation approach to zero.
5. According to Figures 9 to 11, the ranges P and B were shown from the values of the ratio of circulation to evaporation rates are related to the maximum exergy and cooling efficiencies, respectively, and their values are between 1.93 and 2.80 and between 8.50 and 13.15, respectively. So, the optimum range for the above ratio is from 1.93 to 13.15 (from the minimum of range of P to the maximum of range B). On the other hand, based on the only energy (thermal) perspective for the aspen wood-based pad, the ratio of circulation to evaporation rates with a range from 9.5 to 10.5 was suggested [1].
6. Figure 13 is a unique result that shows the exergy efficiency in terms of cooling efficiency for different air velocities passing through the pad. The positions of maximum exergy efficiency are different from the maximum cooling efficiency and with increasing air velocity, the maximum exergy efficiency increases and the maximum cooling efficiency decreases. It should be noted that this curve is the result of dynamic circulation, which is inherently different from static circulation and requires further research for static circulation, which has more uses.

Nomenclature

c.t.c.	Control time cycle
Ex	Exergy (kJ)
m	Mass (kg)
P	Pressure (Pa)
p.o.t.	Pump on-time
Q	Heat transfer (kW)
T	Temperature (°C)
y	Mole fraction
Greek Symbol	
φ	Relative humidity (%)
ω	Absolute humidity (kg vapor/kg dry air)
ϕ_0	The ratio of the water vapor pressure of the unsaturated atmospheric air to the saturated

	water vapor pressure at the dead state temperature
ψ	The ratio of the water vapor pressure of the unsaturated atmospheric air to the saturated water vapor pressure at the dead state temperature
	specific exergy (kJ/kg)

Subscripts

a	Air
chem	Chemical
dest	Destroyed
g	Water vapor saturation
in	Inlet
o	Base
out	Outlet
phy	Physical
Q	Heat transfer
s	Saturated
thermal	Thermal
v	Water vapor in ambient condition (dead state)
w	Water
wb	Wet bulb

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