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Experimental Investigation on the Cooling Performance of a Counterflow Dew Point Evaporative Cooler

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Abstract

In this study, an innovative indirect evaporative cooler is presented and experimentally analysed. Although there are many evaporative cooler designs, this study proposes a unique M-cycle type evaporative cooler with newly designed heat exchanger to enhance the cooling performance of indirect evaporative coolers. The prototype was assembled for the experimental investigation of the system and tested in the laboratory environment under the conditions of constant air flow rate of 350 m^3/h, circulating water temperatures of 15 °C and 20 °C, inlet air temperatures of 25 °C and 30 °C and lastly inlet air humidity of 9g/kg and 13g/kg, respectively. Based on the data obtained during the experiments, calculations were performed for the cooling capacity, cooling efficiency, energy efficiency and exergy efficiency of the system for each case. The findings showed that the highest wet bulb effectiveness, dew point effectiveness and EER of evaporative cooler were found to be 0.91, 0.62, and 0.77, respectively.

Nomenclature		*Subscripts*	
c_p	specific heat, J/g.°C	a	air
e	exergy, J/kg	da	dry air
$\dot{m}$	mass flow rate, kg/s	f	air supply
T	temperature, °C	db	dry bulb
I	exergy destruction, W	wb	wet bulb
i	enthalpy, J/kg	dp	dew point
P	consumed power, W	t	total
R	gas constant kJ/kgK	th	thermal
ρ	density, kg/m^3	me	mechanical
$\dot{Q}$	cooling capacity, W	ch	chemical
W	humidity ratio, g/kg	p	*process*
$\dot{V}$	volume flow rate, m^3/s	w	water
φ	relative humidty	o	resricted dead state
ε	effectiveness	oo	dead state

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Abbreviations	
DEC	Direct evaporative cooling
IEC	Indirect evaporative cooling
EER	Energy efficiency ratio

1. INTRODUCTION

The ever-increasing use of technology and the desire to live in more comfortable environments substantially increase the use of air conditioning systems around the world. Consequently, excessive use of air conditioning systems is responsible for approximately 50% of the energy consumption in the building sector, which has a significant share of 30%-40% energy consumption [1]. Therefore, it is now more crucial to attempt to develop and use more efficient and environmentally friendly processes in air conditioning sector ever before. To reduce emissions from cooling processes and protect the ecosystem, high-energy efficiency “Green Cooling” technologies, which uses natural fluids such as water and air, should be widespread [2–4].

There are great quanta of studies have been performed in the existing literature showing that evaporative coolers are highly energy efficient and very environmentally friendly systems compared to conventional mechanical vapor compression cooling systems [5–6]. Although evaporative coolers are used in residential air conditioning, they can also be effectively used in foundries, livestock farms and greenhouses that needed moist air [7]. Evaporative cooling systems have simple principle based on the evaporation of water introduced into the air. In these systems, the latent heat required for the evaporation of water is taken from the sensible heat of the air. Due to heat and mass transfer, that decrease the air temperature, the cooling process is performed. Structurally, evaporative cooling systems are divided into two types as direct evaporative cooling (DEC) and indirect evaporative cooling (IEC) according to whether the product air is exposed to water or not [1].

Since the air is directly cooled in direct evaporative coolers, wet bulb effectiveness is usually high (80%-90%) in such systems [8]. Although the efficiency of DEC systems is relatively high, their implementations are limited especially in humid areas. They are less preferred due to reasons such as increasing the ambient humidity, causing corrosion of the materials and unsuccessfully meeting the human comfort conditions [9]. However, it is suitable for use in arid climatic regions as it provides comfort by humidifying the air [3]. Indirect evaporative coolers, on the other hand, can provide air-cooling without adding moisture to the product air. Unlike DEC systems, IEC systems have secondary channels and separate the air flow from the water flow. This advantage allows IEC systems to be used in humid areas. However, wet bulb efficiencies are relatively lower (55%-65%) than DEC systems [8]. In this context, to increase the efficiency of IEC systems and eliminate this drawback, a new type of heat exchanger design was introduced by Maisotsenko [10,11]. This design can cool the product air at levels close to the dew point below the wet bulb temperature, and known as M-cycle [12–14].

There are many studies have been conducted on M-cycle evaporative coolers. To exemplify a few, Zhao et al. [15] introduced an M-cycle based counter-flow heat and mass exchanger. In this design, part of the product air is directed to the wet channels through the holes at the end of the flow channels. It was observed that the cooling efficiency was substantially improved. The wet bulb effectiveness and the dew point effectiveness of the system were reached 1.3 and 0.9, respectively. Also, in another study, they presented a performance comparison of M-cycle type cross and counter flow indirect evaporative coolers. The results showed that the counter-flow system yielded 20% more cooling capacity, 15% higher wet bulb effectiveness, and 23% higher dew point effectiveness compared to the cross-flow system [16]. Additionally, a feasibility study of this proposed system was conducted for different regions of China and England. Because of these studies, it was stated that the M-cycle indirect evaporative cooling system can be used in regions where the relative humidity level is below 70% [17–18]. Zhan et al. [19] compared an M-cycle cross-flow cooler with a conventional cross-flow indirect evaporative cooler in a numerical study. It was proved that the wet bulb effectiveness and the cooling capacity of the M-cycle system were 16.7% and 15.7% higher, respectively. Rogdakis et al. [20] conducted a theoretical and experimental study to evaluate the performance of an M-cycle indirect evaporative cooler under the prevailing climatic conditions of Greece. The findings revealed that the system with a wet bulb effectiveness between 97% and 115% can be used in most of the cities of Greece. Riangvilaikul et al. [21] presented changes in the cooling efficiency of an evaporative cooler system according to different temperatures and humidity values in their numerical study. For 35 °C of ambient air temperature 26.4 g/kg of humidity level, they found the wet bulb effectiveness as 1.09 and the dew point effectiveness as 0.86 and confirmed their results experimentally. Lei et al. [22] performed the exergy efficiency analysis of a counter-flow evaporative cooler operating on the M-cycle principle. Findings demonstrated that the exergy efficiency was 21% for 1 m duct length and 0.01 m duct width, 33.5 °C air inlet temperature, and 8.244-g/kg-specific humidity. Aydin et al. [23] evaluated the use of evaporative coolers with dryers in regions where the ambient relative humidity is more than 50%. It was decided that the system with the dryer offers 40% a more specific cooling capacity than the system without a dryer.

This work introduces and experimentally investigates a new type of M-cycle evaporative cooler. The system is unique in terms of its newly designed heat exchanger having M cycle applied to the system. The proposed evaporative cooler was tested in a laboratory environment, and wet bulb effectiveness, dew point effectiveness and EER of the system performance were determined. In this context, this study is thought to be important in terms of its potential contribution to the current literature and efforts in the field of evaporative cooling technologies.

2. DESCRIPTION OF THE DEW POINT EVAPORATIVE COOLER

Evaporative cooling has mainly two configurations as direct evaporative cooler (DEC) and indirect evaporative cooler (IEC). DEC systems cool the air entering the system via pads, which are kept constant wet (Figure 1). However, since the air is in direct contact with water, the humidity of the incoming air increases significantly (70%-90%) [24]. However, IEC systems have separate water and air channels. Heat transfer arises through the surface between these two channels. Since air and water flows do not come into direct contact, cooling is performed by means of latent heat and without a significant increase in the humidity of the product air (Figure 2).

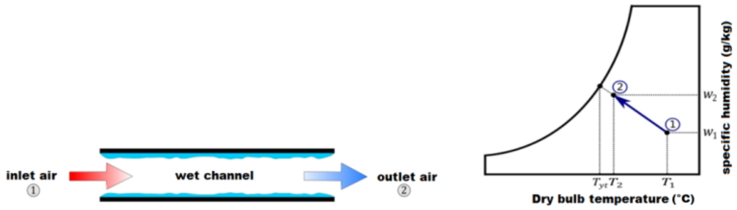

Figure 1. *Direct evaporative cooling (DEC) [25]*

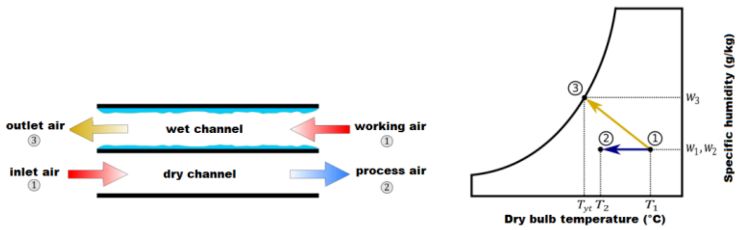

Figure2. *Indirect evaporative cooling (IDEC) [25]*

In this study, an evaporative cooler with an improved M-cycle heat exchanger design that can effectively cool with an insignificant increase in humidity has been introduced [10]. M-cycle evaporative coolers also have two separate dry and wet channels. However, unlike IEC systems, part of the product air entering the system from the dry channel is transferred at the end of each dry channel to the wet channels (working air) (see Figure 3). The process air loses its heat to the end of the dry channel and is transferred to the wet channel, thereby increasing the cooling efficiency of the system. Working air flowing in the opposite direction to the inlet air causes the water on the surface of the wet channel to evaporate, allowing heat to be absorbed from the channel walls. While the working air flowing in the wet channel is exhausted from the system as exhaust air, the inlet air that is not directed into the wet channel is used as a product air for cooling. As a result, the system can achieve 85% dew point efficiencies, thanks to the working air that passes through the wet channel after pre-cooling [15].

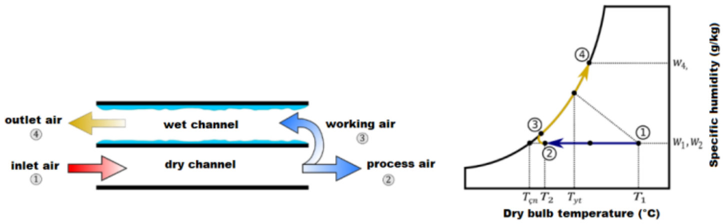

Figure3. *M-cycle indirect evaporative cooler [25]*

2. EXPERIMENTAL SETUP

The prototype of the proposed system assembled is shown in Figure 4. Holes are drilled at the end of each channel to introduce the working air from the dry channel to the wet channel (see Figure 4a). The channels are stacked on top of each other thanks to the bends on the edges of the wet and dry channels (see Figure 4b). Additionally, plates with equilateral triangle geometry were placed between each wet and dry channels to increase cooling efficiency and the heat exchanger was fitted and sealed. Then, the heat exchanger was placed inside the construction so that it could stand in the right position and the necessary fittings were retained. A fan was incorporated into the system, ensuring that the inlet air is transferred to the dry channels. Additionally, a blower fan provides that fraction of the inlet air (working air) in the dry channels passes through the holes to the wet duct and is discharged from the heat exchanger as exhaust air. The pump in the system maintains the circulation of the cooling water from the reservoir to the wet channels. A 25-L capacity reservoir is used to collect the circulating water. A completely assembled and insulated system, with all mechanical and electrical connections, is seen in Figure 4b.

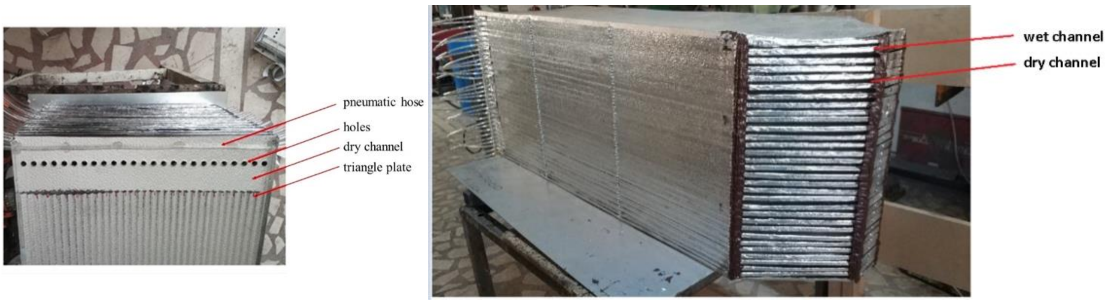

Figure 4. *(a) Dry and wet channels*

Figure 4. *(b) Assembly and test setup of the proposed system*

The wet and dry channels of the system and flows inside the channels are shown in the schematic diagram of the heat exchanger (see Figure 5). As can be seen in the figure, the air entering from the lower left-hand side of the heat exchanger moves along the dry channel and fraction of the air passes into the wet channels at the end of each channel (working channel). The air passing through the wet duct, which has a continuous water flow from above, flows in the opposite direction to the inlet air and exits as exhaust air from the lower right-hand side of the heat exchanger. The process air that enters the system and does not pass into the wet duct is blown to the air-conditioned area (product air).

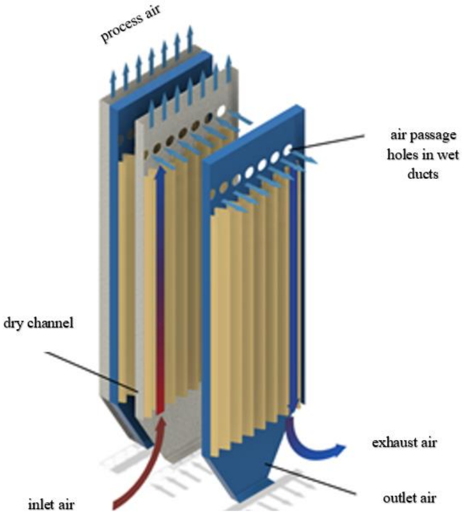

Figure 5. *Schematic view of the M-cycle heat exchanger*

The proposed evaporative cooler has a fan with a maximum flow rate of 520 m³/h for process air inlet and a blower with a flow rate of 410 m³/h and 80 W power for exhaust air. A pump with a flow rate of 3 m³/h was also used for water circulation. Additionally, ENDA ESHT-102-W-50 temperature and humidity sensor with 1 °C temperature and 2% humidity sensitivity is used to measure the temperature and humidity values of the process, product and exhaust air. A VFA float type flowmeter with a measuring range of 100–3000 l/h was used to measure the flow of water. Air velocities entering and leaving the cooler were measured manually using the CEM DT-619 anemometer with ±0.20 m/s sensitivity. All related data were recorded at a sampling rate of 10s by Novus data logger with 8 analog and digital channels each. A direct link between data logger and PC was provided to save and export data on Excel spreadsheet. The technical details of the components and measuring devices used are presented in Table 1.

Table 1. *Technical specifications of system components and measuring devices*

Blower fan (exhaust air)	*Power*	*80 W*
	Voltage	*220 V*
	Revolutions Per Minute	*2325*
	Max. flow rate	*410 m³/h*
Fan (for inlet air)	*Power*	*210 W*
	Voltage	*220 V*
	Revolutions Per Minute	*2800*
	Max. flow rate	*520 m³/h*
Circulation pump	*Power*	*750 W*
	Voltage	*220 V*
	Max. flow rate	*3 m³/h*
Anemometer	*Range*	*0.4–30 m/s*
	Accuracy	*± 0.20 m/s*
Hum. and Temp. transmitter	*Temperature*	*-40 °C to +125 °C*
	Accuracy	*± 1 (0-+70 °C)* *±0.5 (20-+40 °C)* *2% RH*
Datalogger	*# of channels*	*8 digital, 8 analog*
	Analog input types	*Thermocouples, V, mV, mA, Pt100*

3.PERFORMANCE ANALYSIS

A numerical model suitably tailored to the configuration of the proposed system was adapted to simulate the thermal performance of the system. For the sake of simplicity, several assumptions were considered during the thermodynamic analysis of the system as follows;

• The heat is transferred vertically to the partition plate,
• Air flow rate is constant throughout the duct,
• There is always water on the wet surfaces of the channels,
• Air is considered as incompressible since the Mach number of the air flow rate is less than 0.3,
• As the thickness of the partition plate is low, the temperature difference between the dry and wet surfaces of the partition plate can be neglected.

4.ENERGY ANALYSIS

To evaluate the system performance of M-cycle evaporative coolers, wet bulb and dew point effectiveness values are examined. These efficiency values are calculated according to the temperature values of the air entering and leaving the system. The wet bulb effectiveness, as seen in Equation 1, is the ratio of the dry bulb temperature difference in the inlet and outlet air to the inlet dry bulb temperature and the inlet wet bulb temperature difference [22].

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

The dew point effectiveness, on the other hand, is the ratio of the dry bulb temperature difference in the inlet and outlet air to the inlet dry bulb temperature and the inlet dew bulb temperature difference, as seen in Equation 2.

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

The cooling capacity of the evaporative cooler is calculated using the following equation. Here, $c_{p,a}$ is the specific heat of the inlet air, ρ_a is the density of the air, $\dot{V}$ is the volumetric flow rate of the air, T_{in} is the temperature of the inlet air and T_{out} is the outlet temperature of the process air [22].

$$\dot{Q}_{cooling} = c_{p,a}\rho_a\dot{V}(T_{in-}T_{out}) \tag{3}$$

The Energy Efficiency Ratio (EER) of the evaporative cooler is calculated using the following equation. EER is found by the ratio of the cooling capacity to the total power consumption of the cooler. It is preferred instead of COP in cooling systems, which provides insight about how much cooling capacity could be obtained from a specific amount of energy used. EER requires the calculation of electrical power consumption by fan and water pump in the system [22].

$$EER = \frac{\dot{Q}_{cooling}}{P_{total}} \tag{4}$$

Where P_{total} is the total energy consumed instantly by each device in the system and it is determined by measurement.

5.EXERGY ANALYSIS

The exergy in the air conditioning process, which reaches thermal, mechanical and chemical equilibrium under ambient conditions, can be written as follows [22]

$$e_t = e_{th} + e_{me} + e_{ch} \tag{5}$$

Air and water are two types of fluids flowing across the evaporative cooler. Wet air can be considered a mixture of ideal gas consisting of dry air and water vapor. Then, the exergy of moist air and water can be written as [22]

$$e_a = (c_{da} + Wc_v)T_0\left(\frac{T}{T_0} - 1 - ln\frac{T}{T_0}\right) + (1 + 1.608W)R_aT_0ln\frac{P}{P_0} + R_aT_0\left[(1 + 1.608W)ln\frac{1+1.608W_{00}}{1+1.608W} + 1 + 1.608Wln\frac{W}{W_{00}}\right] \tag{6}$$

$$e_w = [i_w(T) - i_w(T_0)] - T_0[s_w(T) - s_w(T_0)] - R_vT_0ln\varphi_0 \tag{7}$$

The definition of the dead state condition is pivotal for exergy analysis. Generally, a stable atmospheric conditions are chosen as reference. However, air that is not saturated with moisture under atmospheric conditions still has useful energy. Therefore, the dead state is defined as the saturated outside air in the study. The exergy of this is, then, expressed as [22]

$$\left(m_1e_{1,a} + m_{w,i}e_{w,i}\right) = \left(m_fe_{2,a} + m_3e_{3,a}\right) + I \tag{8}$$

Where I is the exergy destruction, which is a measure of resource degradation and specifies where in system destruction occurs. The exergy destruction describes the lost work potential and is also called irreversibility or exergy loss that occurs mainly due to air leakage, pressure loss and flow friction in evaporative cooling systems.

In the heat and mass transfer process, the moisture added to the air is considered as the amount of water inlet. The exergy efficiency of evaporative cooler is defined as the ratio between the useful exergy of obtained cooling effect and overall exergy used to drive the evaporative cooling system. The second law, exergy, is an important criterion for evaluating cooling performance, and exergy efficiency is defined as [22]

$$\eta_e = 1 - \frac{I}{(m_1e_{1,a} + m_{w,i}\,e_{w,i})} \tag{9}$$

Uncertainty of the experimental results

The relative uncertainty in the efficiency and cooling capacity calculations based on the temperature and humidity values obtained from the tests can be by [26]

$$\frac{\Delta x}{y} = \sqrt{\sum_i \left(\frac{\partial y}{\partial xi}, \frac{\Delta xi}{y}\right)^2} \tag{10}$$

Where x and y refer to the directly measured and calculated parameters. It was found that the relative uncertainty of wet bulb thermometer and dew point effectiveness was 7%, while the relative uncertainty of the cooling capacity was 9.1%. It is noted that the main parameter affecting the efficiency and cooling uncertainty is the sensitivity of the temperature sensors used in the measurements. Utilisation of high sensitivity equipment in assessment and evaluation would substantially reduce the uncertainty values attained in this study. However, when the results were compared with similar studies in the literature, the uncertainties were found reasonable [26].

6.EXPERIMENTAL RESULTS AND DISCUSSION

In this study, an M-cycle counter-flow evaporative cooler was experimentally tested in the Low Carbon Technology Laboratories of Science and Technology Research and Application Centre (BITAM) of Necmettin Erbakan University. The tests were conducted at water temperatures of 15 °C and 20 °C, air inlet temperatures of 25 °C and 30 °C and air inlet humidity of 9g/kg and 13g/kg, respectively. The corresponding data on product and exhaust air were recorded. Wet bulb effectiveness, dew point effectiveness, EER, cooling capacity and exergy values related to system performance were calculated using the obtained data and evaluations were performed on the basis of these calculations. Therefore, the effect of cooling water and inlet air parameters on the performance of an M-cycle evaporative cooler is demonstrated.

In this section, the findings obtained from the laboratory tests of the proposed evaporative cooler are evaluated. Temperature and humidity data of product air and exhaust air were recorded in 8 different tests performed at different water temperatures, air inlet temperatures and humidity values, and performance analyses of the system were conducted in line with the recorded data. Table 2 demonstrates the test conditions.

Table 2. *Experimental test conditions*

	Inlet air flow rate (m^3/h)	Water temperature (°C)	Inlet air temperature (°C)	Inlet air humidity ratio (g/kg)
1. Exp.	350	15	30	9
2. Exp.	350	15	30	13
3. Exp.	350	15	25	9
4. Exp.	350	15	25	13
5. Exp.	350	20	30	9
6. Exp.	350	20	30	13
7. Exp.	350	20	25	9
8. Exp.	350	20	25	13

6.1. 15 °C of Water Temperature

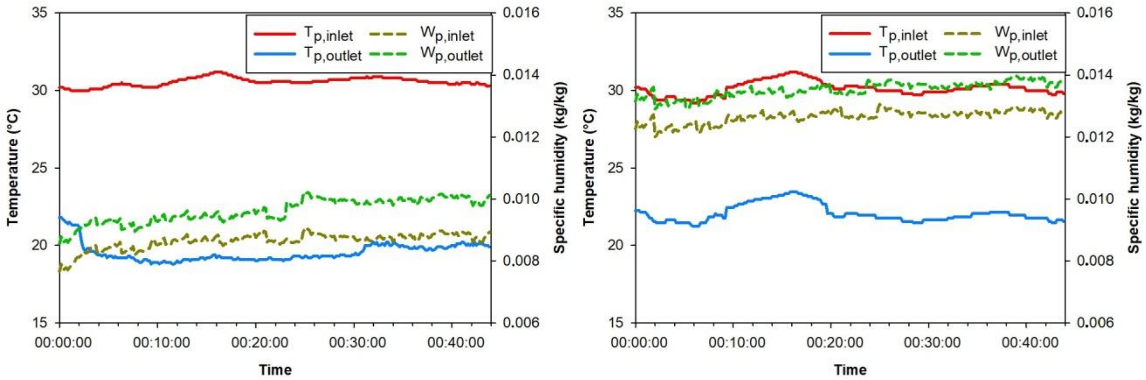

Figure 6. *Findings for 15 °C of water and 30 °C of inlet air temperatures at (a) 9 g/kg and (b) 13 g/kg of inlet air humidity levels*

Figure 6 demonstrates the test results conducted for 9g/kg and 13g/kg moisture levels when the water temperature was 15 °C and the air inlet temperature was 30 °C. While the humidity of the inlet air was 9g/kg (Figure 6a.), the inlet air temperature (30 °C) decreased by about 9 °C at the outlet and the humidity increased by about 1g/kg. While the inlet air humidity was 13g/kg (Figure 6b), the air was cooled by 7 °C down to 23 °C. Similarly, the humidity level increased by 1 g/kg and departed the system as approximately 13.5 g/kg.

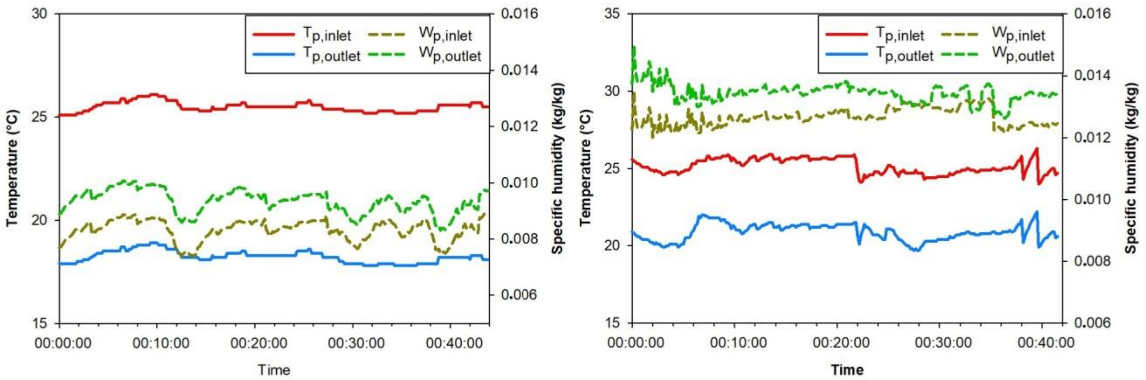

Figure 7. *Findings for 15 °C of water and 25 °C of inlet air temperatures at (a) 9 g/kg and (b) 13 g/kg of inlet air humidity levels*

When the water temperature was still 15 °C and the air inlet temperature was set to 25 °C, two tests were conducted for the humidity of 9 g/kg and 13 g/kg. The values of the temperature and humidity at the inlet-outlet of the system are presented in Figure 7a and 7b. In the test performed for 9g/kg air inlet humidity, the inlet air at 25 °C was cooled by approximately 6.5 °C, while it was down by approximately 5 °C for 13g/kg air inlet humidity. Moisture levels increased by approximately 1g/kg in both tests. It is seen that the air was cooled more at 30 °C air inlet temperature (Figure 6) then 25 °C air inlet temperature (Figure 7).

6.2. 20 *°C* of Water Temparature

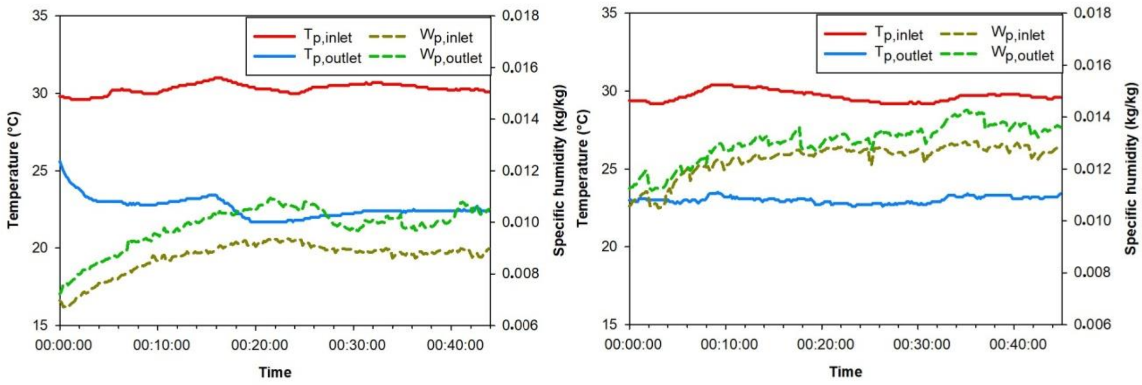

Figure 8. *Findings for 20 °C of water and 30 °C of inlet air temperatures at (a) 9 g/kg and (b) 13 g/kg of inlet air humidity levels*

The results obtained in the tests when the circulating water temperature in the system was 20 °C and the inlet air temperature was 30 °C are shown in Figure 8a and 8b, respectively. While the water temperature was 20 °C, the air inlet temperature was 30 °C, the air inlet humidity was 9 g/kg, the air was humidified by approximately 1g/kg and its temperature was reduced by around 8 °C (see Figure 8a). The air was cooled by 6.5 °C by humidifying approximately 1g/kg when the inlet air humidity level was 13 g/kg.

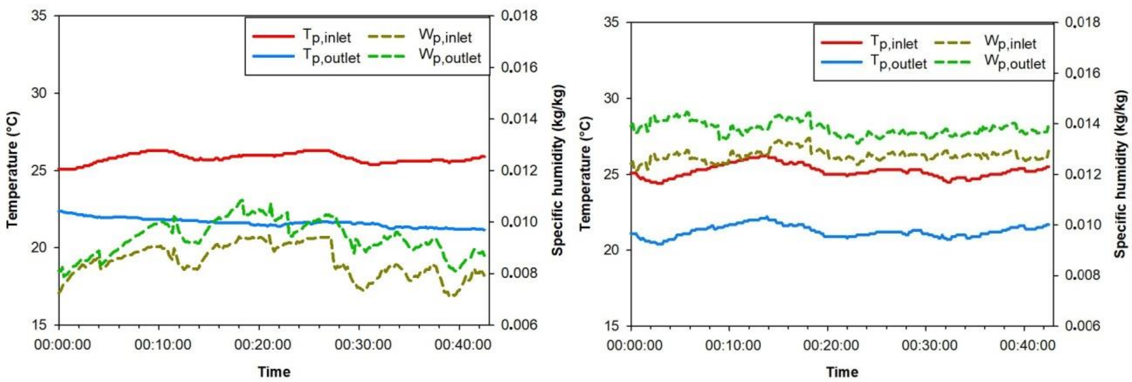

Figure 9. *Findings for 20 °C of water and 25 °C of inlet air temperatures at (a) 9 g/kg and (b) 13 g/kg of inlet air humidity levels*

Figure 9 demonstrates the findings when the water temperature was 20 °C, air inlet temperature was 25 °C and inlet air humidity levels were 9 g/kg and 13g/kg, respectively. It has been determined that when the water temperature was 15 °C, compared to the 20 °C water temperature, the system could perform better as shown in Figure 9. While the air inlet temperature and air humidity were 25 °C and 9g/kg and 13g/kg, the air was chilled by approximately 4.2 °C and 3.5 °C, respectively. In some cases, extra energy could be needed to supply 15 °C water, which contributes to the increase in cooling capacity.

6.3. Wet Bulb Temperature and Dew Point Effectiveness

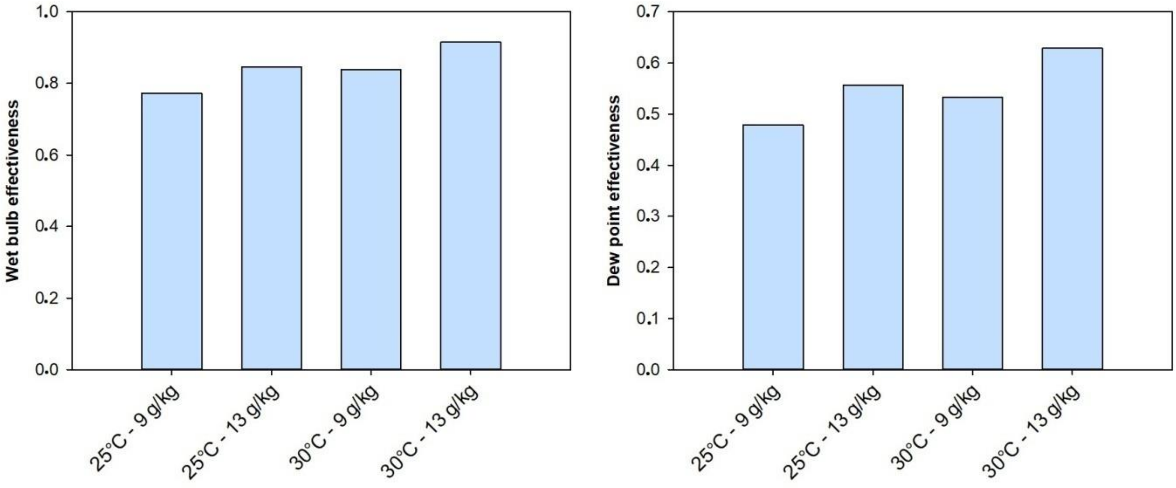

Figure 10. *(a) Wet bulb and (b) dew point effectiveness at 15 °C water temperature*

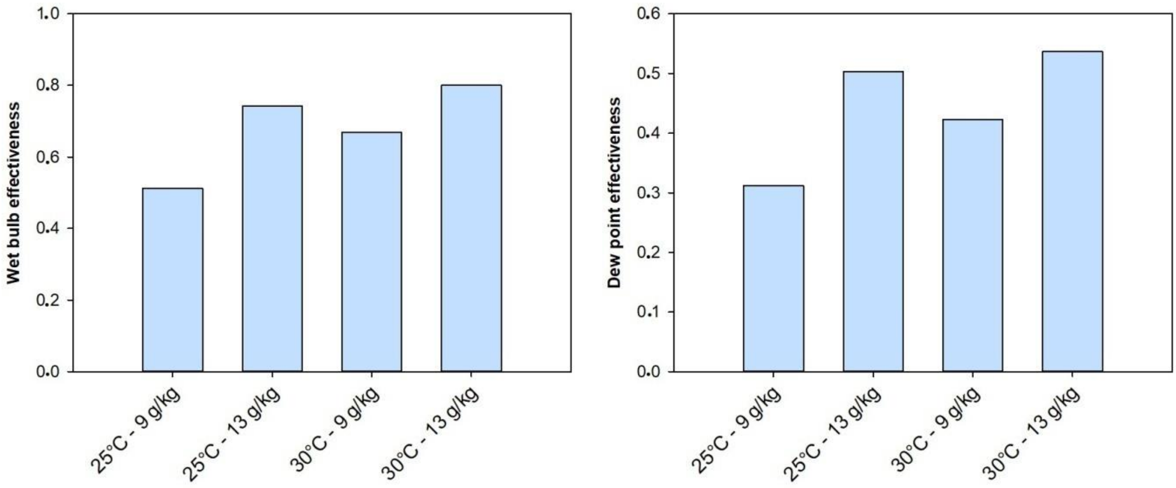

Figure 11. *(a) Wet bulb and (b) dew point effectiveness at 20 °C water temperature*

Dew point and wet bulb effectiveness are decisive parameters in the performance evaluation of evaporative cooling systems. The wet bulb and dew point effectiveness depend on the temperature of the outlet air (product air) and inlet air, and humidity values as expressed in Eqns. 1 and 2. The higher the wet bulb and dew point effectiveness, the closer the outlet air temperature approaches the inlet air wet bulb and dew point temperature. As can be seen in Figures 10 and 11, it is shown that when wet bulb and dew point effectiveness increase, the temperature and humidity of the inlet air increase as well. In contrast, when the water temperature increases, it is noticed that the wet bulb and dew point effectiveness decrease.

While the water temperature was 15 °C, the inlet air temperature was 30° C and the specific humidity was 13 g/kg, the highest wet-bulb and dew point effectiveness values were achieved as 0.91 and 0.62, respectively. While the water temperature was 20 °C, the inlet air temperature was 30 °C and the specific humidity was 13 g/kg, the highest wet bulb and dew point effectiveness were attained as 0.80 and 0.53, respectively. As shown, the cooling effectiveness is highly affected by the inlet air temperature and humidity values, while the water temperature has relatively less impact on the cooling effectiveness.

6.4. Cooling Capacity and EER

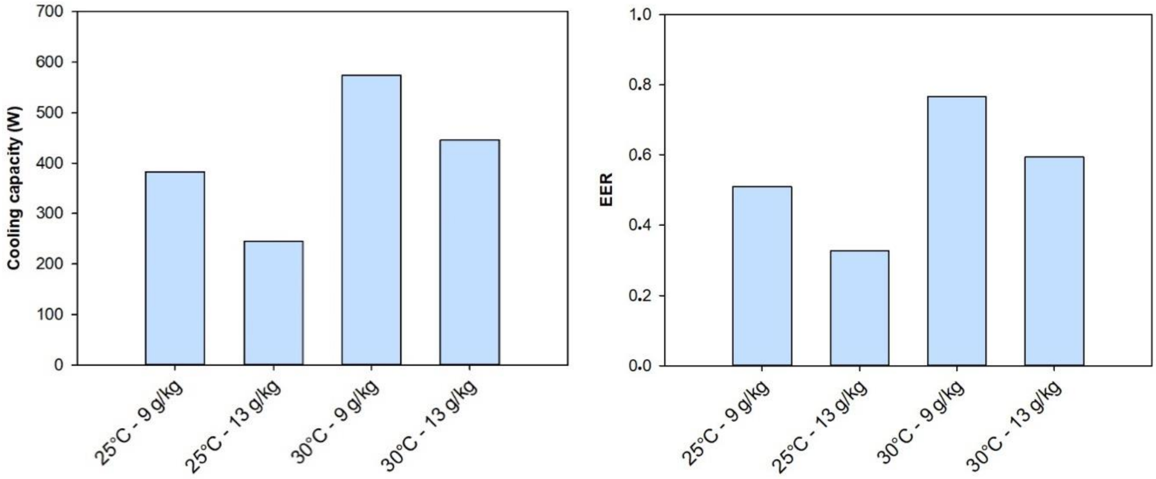

Figure 12. *(a) Cooling capacity and (b) EER at 15 °C water temperature*

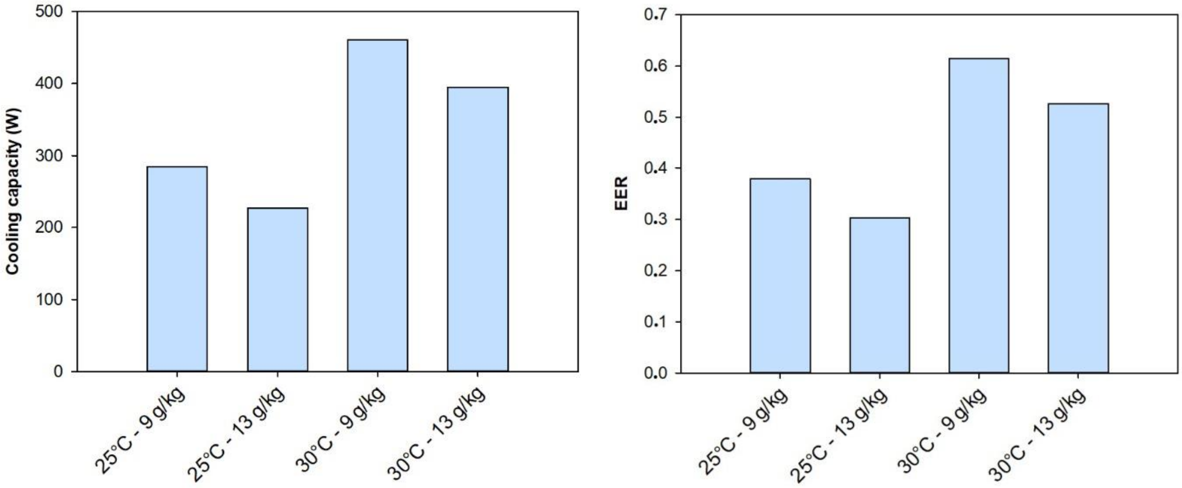

Figure 13. *(a) Cooling capacity and (b) EER at 20 °C water temperature*

In evaporative cooling systems, cooling capacity and EER are also essential in addition to the cooling efficiency. The cooling capacity of an evaporative cooler should sufficiently encounter the heat gains in the air-conditioned space. EER, on the other hand, indicates how efficiently the system works. For 15 °C and 20 °C water temperatures, the highest cooling capacity was achieved as 574.5 and 460.8 W, and highest EER was attained as 77.6% and 61.4% at inlet air temperature of 30 °C and specific humidity of 9 g/kg. However, the lowest cooling capacities were recorded as 245.3 and 227.4 W, and the lowest EERs were 32.7% and 30.3% at 25 °C of inlet air temperature and 13 g/kg of specific humidity. It can be deduced that the cooling capacity and EER decrease as the water temperature and specific humidity increase. Yet, as the inlet air temperature increases, the cooling capacity and EER values increase. However, note that the EER of the system will decrease if extra energy must cool down the water temperature circulating across the system.

6.5. Exergy Efficiencies and Destruction

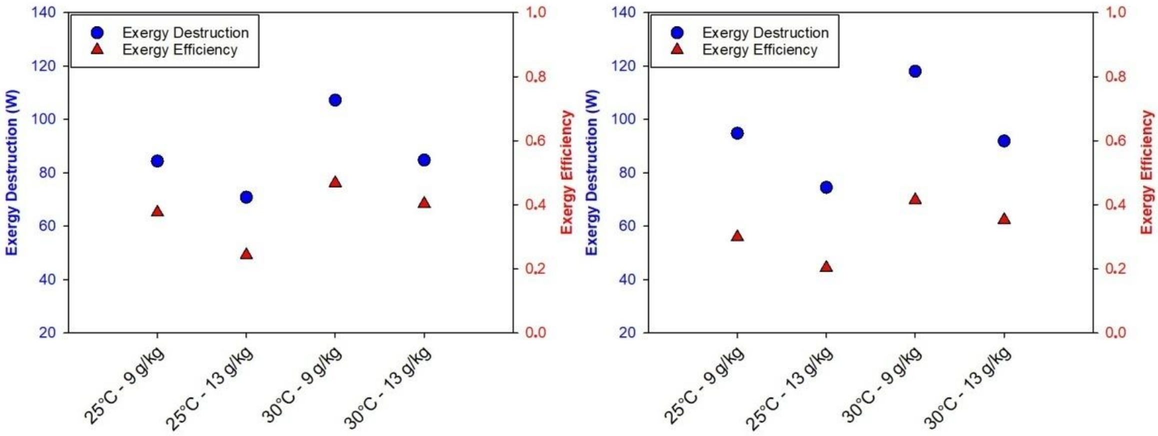

Figure 14. *Exergy efficiencies and exergy destruction at (a) 15 °C and (b) 20 °C water temperatures*

The exergy analyses in line with the data obtained from the tests performed under different water temperature and air inlet conditions are shown in Figure 14. Findings reveal that the exergy efficiency and exergy destruction increases with the increasing air inlet temperature. Adversely, exergy efficiency and destruction decrease as the inlet humidity increases. The highest exergy efficiency was found as 0.46 at air inlet temperature of 30 °C and air inlet humidity of 9 g/kg, while the lowest exergy efficiency was 0.20 at air inlet temperature of 25 °C and inlet humidity of 13 g/kg air. Corresponding exergy destruction was 110 W and 75 W, respectively.

The energetic and exergetic performance of the investigated M cycle evaporative cooler could be further improved by increasing the distance travelled by process air inside the heat exchanger, which would increase the cooling capacity. However, the pressure drop due to air friction is a drawback which may be overcome through oval channels.

7. CONCLUSION

In this study, an experimental investigation on the cooling performance of an innovative M-cycle evaporative cooler was conducted. Series of tests were performed at different water temperatures (15 °C and 20 °C) and different air inlet conditions (inlet temperature of 25 °C and 30 °C and inlet humidity of 9g/kg and 13g/kg). The concluding remarks outlined are as follows:

- It was noted that the water temperature has a significant impact on the wet bulb and dew point effectiveness, that can seriously alter the product air outlet temperature by approximately 2–3 °C.
- The inlet air temperature and inlet air humidity has the reverse effect on the wet bulb and dew point effectiveness. As such, the wet bulb and dew point effectiveness increase with the increasing inlet air temperature and decrease with the increasing inlet air humidity.
- It was observed that the air inlet humidity increased by about 1 g/kg. This humidity increase is minimal comparing to the humidity increase in conventional direct evaporative coolers.
- The highest wet bulb and dew point efficiencies obtained from the system were attained at the air inlet temperature of 30 °C and air inlet humidity of 13 g/kg. The highest wet-bulb effectiveness at water temperatures of 15 °C and 20 °C was found to be 91.5% and 80%, respectively, while the highest dew point effectiveness was found to be 62.8% and 53.6%, respectively.
- The highest EER value was achieved as 76.6% at a water temperature of 15 °C, air inlet temperature of 30 °C and air intake humidity of 9 g/kg.
- The highest exergy efficiency was achieved as 46.8% at 15 °C water temperature, 30 °C air inlet temperature, and 9 g/kg air inlet humidity. However, the lowest exergy efficiency was attained as 20.3% at a water temperature of 20 °C, air inlet temperature of 25 °C and air intake humidity of 13 g/kg.

It was also deduced that the performance of the developed M-cycle cooler can be further improved substantially through;

- sealing each layer of the heat exchanger separately during assembly,
- constructing an outer case, preventing the system being exposed to the external environment,
- preserving water temperature by using water storage without exposing the water to the ambient temperature
- and finally utilising blower fans with lower capacities.

It is inferred that the proposed cooling unit can be used in hot, dry and even humid climates. In humid climates, however, a dehumidifier must be integrated into the system so the air humidity level can be adjusted to human comfort conditions.

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