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Chapter

Advancements in Indirect Evaporative Cooling Systems through Novel Operational Configuration

Muhammad Ahmad Jamil, Muhammad Wakil Shahzad, Ben Bin Xu, Muhammad Waqar Ashraf, Kim Choon Ng, Nida Imtiaz and Haseeb Yaqoob

Abstract

Rising global temperature has triggered the cooling demand in the last three decades with growing predictions for the future. The use of conventional energy-intensive and high global warming chemical-based cooling systems is working in a loop, increasing the global warming rate, emissions, and cooling system inventory. Therefore, the development of an innovative cooling system with high energy efficiency, low monetary cost, and environmentally sustainable. The indirect evaporative cooling-based systems have shown potential to serve the purpose because of low energy consumption, absence of energy, and cost-intensive equipment like compressors and water-based operation. A novel indirect evaporative cooler based on an innovative operational configuration is proposed, fabricated, and tested experimentally. The Proposed system has several advancements compared to the conventional indirect evaporative coolers like high operational reliability, low maintenance, and better control of the processes in the system. The study shows that the proposed system can achieve a temperature drop of as high as 14°C. The maximum cooling capacity of the system is calculated as 110 W, and the cooling performance index of 28. The performance of the cooler improves with increasing outdoor air temperature which makes it suitable for diverse climatic conditions. Moreover, the proposed design offers several benefits due to novel operational configurations by addressing limitations in the earlier systems.

Keywords: advanced evaporative cooler, humidity-controlled cooling, cleaner air conditioning, sustainable development

1. Introduction

A remarkable surge in the global energy demand has been seen in the recent past because of an exponential rise in population, urbanization, and economic growth [1].

Moreover, the improving lifestyle is also contributing significantly to the energy consumption to maintain human comfort [2, 3]. The worldwide energy demand is estimated to rise to 850 quadrillions Btu by 2050 from merely 500 quadrillions Btu in 2010 indicating around a 50% rise in 4 decades [4]. Particularly, the situation is alarming for developing countries that will face a tremendous surge of around 71%in the energy demand compared to the developed countries with 18% [5, 6]. Building energy consumption is of critical importance in the overall energy consumption scenario because of the major share and diverse necessary activities [7]. The common among these are human thermal comfort, cleaning, cooking, food preservation, lightening, etc. Despite the wide range of activities taking place in buildings, the overall energy consumption can be classified into 6 different types as shown in **Figure 1** [8, 9]. For commercial and residential buildings, the energy consumption is as follows: heat ventilation and air conditioning 36–40%, lighting 12–20% hot water supply 9–13%, electronics 8–15%, refrigeration 5–7%, cooking 4–5% and others 8–18% [10, 11].

Meanwhile, the demand for air conditioning is also increasing continuously and the global air conditioner inventory is expected to cross 5600 million by 2050 from merely 1600 million units reported in 2016 [12]. The corresponding energy consumption and emission are also expected to surge to 6200-Terawatt hour and 170 Gigatons by 2050 [13]. The use of conventional vapor compression chillers is one of the major reasons for these high energy demands. This is because these systems have very low energy efficiency, which is low efficiency, and involve high global warming potential refrigerants for cooling [14]. Meanwhile, their performance has not seen any considerable improvement in the last 30 years with a stagnant coefficient of performance (COP) of 3–4. This is because of rigid temperature lifts (5 to 7°C) across the evaporator and condenser which require a high input energy compressor [15]. Moreover, high global warming potential chemical refrigerants are used for compression and expansion in the system for cooling with high chances of leakage due to elevated pressure operation. These problems cannot be abandoned in conventional operational schemes. So, an innovative system is required to achieve a breakthrough in the cooling sector [16].

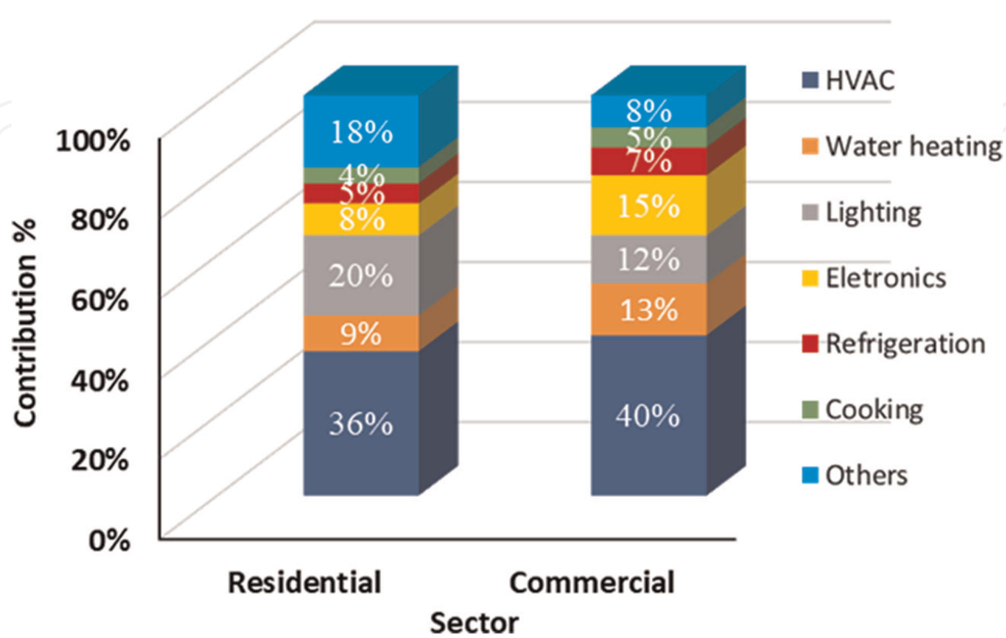

Figure 1.
Building energy consumption distribution [8–10].

One of the lucrative options for the above-mentioned problems emerged is the indirect evaporative cooling system [17]. It uses a water-based cooling mechanism and does not involve any hazardous chemical refrigerant [18]. Moreover, it does not involve any high energy consumption compressor. Rather these systems use fans for air movement through the system and pump for water supply [19, 20]. In these systems, the hot outer air is cooled using cold wet air (water-air mixture) in two different channels. The channels are separated by thermally conductive impermeable walls which only allow heat exchange between the air streams without any moisture exchange. Therefore, these systems produce cold dry air using air and water through evaporation [21, 22].

These systems have been extensively studied by the research community from an experimental and theoretical viewpoint. For instance, multipoint air injection in IECs improved the cooling performance by achieving an additional 2 to 3°C temperature drop by achieving COP as high as 78 for cooling [23]. Likewise, the enhancement of heat transfer plates through protrusion was reported to enhance the IEC performance achieving wet bulb efficiency up to 85% [24, 25]. Chua et al. [26] investigated a felt-assisted IEC with a cross-flow heat exchanger and showed that the cooler can achieve wet bulb efficiency of 90%. Similarly, Duan et al. [27] developed a compact heat exchangers based counter flow IEC. They reported the performance characteristics of the system in terms of wet bulb efficiency up to 107%, cooling capacity up to 8.5 kW, and energy efficiency ratio up to 20. They also improved the system efficiency and energy efficiency ratio by 30% and 40%, respectively by enhancing plates with corrugations [28]. Similarly, finned channel-based systems have also shown promising performance with wet bulb efficiency as118–122% and dewpoint effectiveness of 75–90% [29]. Cui et al. [30] investigated the hybrid air conditioning system with IEC and reported the COP as 14.2 for cooling.

Besides experimental studies, the theoretical analyses of IEC systems have also shown considerable performance improvements through design and operational modifications. For instance, Oh et al. [31] studied the effect of purge configuration on regenerative type IEC performance. They reported the maximum cooling performance at 35% purge ratio with a dew point efficiency of 58% and cooling capacity of 59 W. Similarly, Pandelidis et al. [32] showed that the performance of regenerative IEC systems can be improved through additional perforations, particularly at higher air flow rate ratio > 45%. Riangvilaikul et al. [33] showed that the Polyurethane based IEC system showed promising performance by achieving dewpoint efficiency of 65–87% and the wet bulb efficiency 106–109%. Wang et al. [34] optimized a dewpoint regenerative type IEC based on the entropy principle. The optimal values for velocity, heat exchanger length, channel gap, and airflow rate ratio were reported as 1.0 m/s, 1–1.75 m, 3-5 mm, and 30–40%, respectively. Jradi et al. [35] optimized the dew point cooler for maximum wet bulb efficiency of 112%, and dewpoint efficiency of 78%. The optimal channel gap and heat exchanger length under considered operating conditions were reported as 5 mm and 500 mm, respectively. Similarly, Adam et al. [36] optimized the cross-flow IEC and reported the supply air temperature as 24°C and wet bulb efficiency as 92%. and $T_{PA,o}$ = 24°C. Some other efforts include the study of the effects of condensation in the dry channel [37], wettability enhancement in the wet channel [38], and improving heat transfer characteristics of heat exchangers [39].

The literature review shows that indirect evaporative coolers are a capable substitute for compression cooling systems because of promising performance from temperature drop, efficiency, and COP. Therefore, an innovative indirect evaporative cooler with a novel operational configuration is proposed and tested experimentally.

The proposed system has several advancements like simple construction, separate control for the air-water mixing process, fewer maintenance requirements, and better control of sensible and latent heat transfer processes. The current study analyzes a generic cell of the system which can be used to develop the design metrics for commercial-scale expansion of the proposed idea.

2. Proposed system

2.1 Design concept

The current system consists of the following major components: a heat exchanger (HE), a mixing chamber (MX), fans, a water pump, and nozzles as shown in **Figure 2**. The heat exchanger consists of three identical channels designated for supply air (1 middle channel) and working air (2 side channels). The channels are constructed using a conductive impermeable wall. Spacers are used to maintain the required channel gap to avoid sheet bulging at high air flow rates. The mixing chamber is constructed using acrylic sheets. The chamber is equipped with water spray systems (nozzles, pump, and supplementary water float inlet), and working air manifolds. The sump of the humidifier contains water that is used in a recirculation manner for cooling the working air. As the water is consumed during evaporation, the supplementary water is fed to the system from the reservoir through a float-controlled valve. Besides, the system is also equipped with two axial fans for product air and working air supply. The working cycle of the system is presented below.

In the proposed system the hot outer air undergoes sensible cooling (1 to 2) in the middle (dry) channel of the heat exchanger. The outdoor air flows from the bottom of the channel to the top. An axial fan is employed to maintain the required flow rate and overcome the channel frictional pressure drop. This air stream rejects the heat to the working air in the two side channels flowing in the counter-current direction (from top to bottom). While the working air stream first enters the mixing chamber (as hot outer air) where it is mixed with fine water mist generated using atomizing nozzles. This air-water meshing causes the hot air to release its heat through evaporation and gain high humidity (100%) (1 to 3). This cold humid working air carrying fine water particles enters the heat exchanger and extracts heat from the dry channels. The humidity drops and the temperature rises as the working air moves along the heat exchanger because of latent and sensible heat transfer respectively (3 to 4). The product air (dry and cold) is supplied to the conditioned space and the working air (hot and humid) is discarded to the outer environment.

2.2 Experimental test rig

Based on the above-presented design concept, an experimental test rig was developed as shown in **Figure 3**. Different components of the system are labeled as, A: heat exchanger, B: mixing chamber, C: outer air inlet to heat exchanger, D: product air outlet, E: air inlet to MX, F: working air inlet to HE, G: working air outlet, and H: atomizing nozzles. The heat exchanger was constructed using 0.025 mm thick high thermal conductivity (235 W/m K) Aluminum sheets. The channel gap was maintained at 5 mm using closed foam spaces. The acrylic sheets were used to support the heat exchanger structure and act as adiabatic walls for working air channels. The

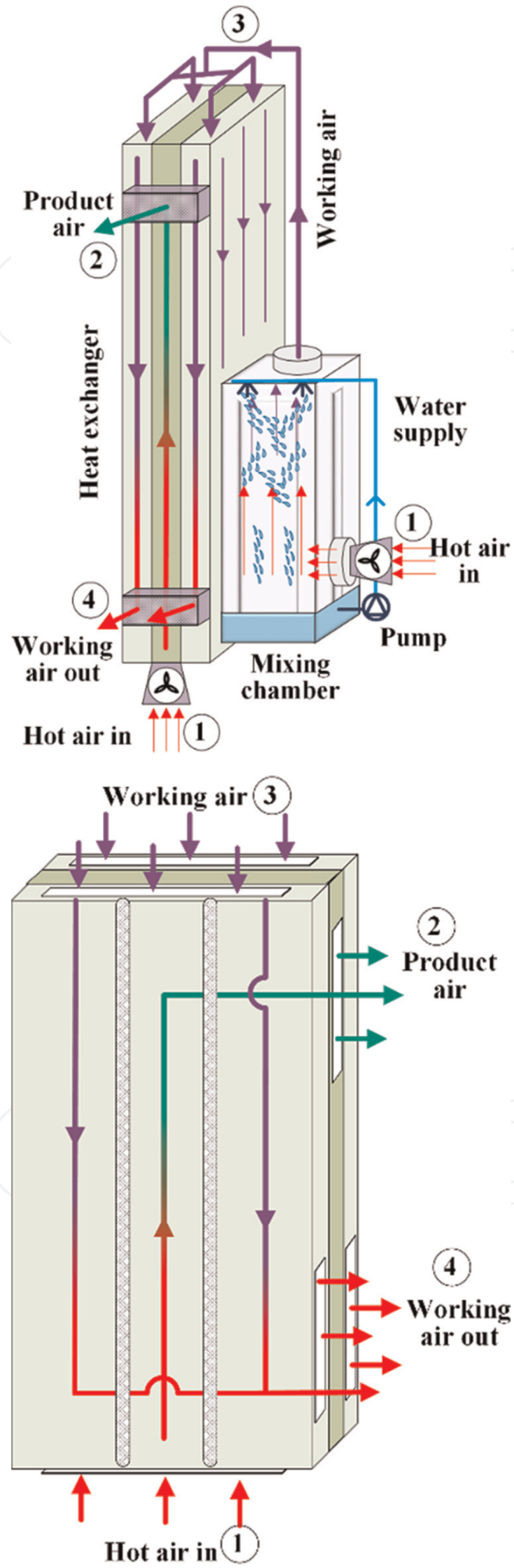

Figure 2.
Schematic diagram of the proposed system.

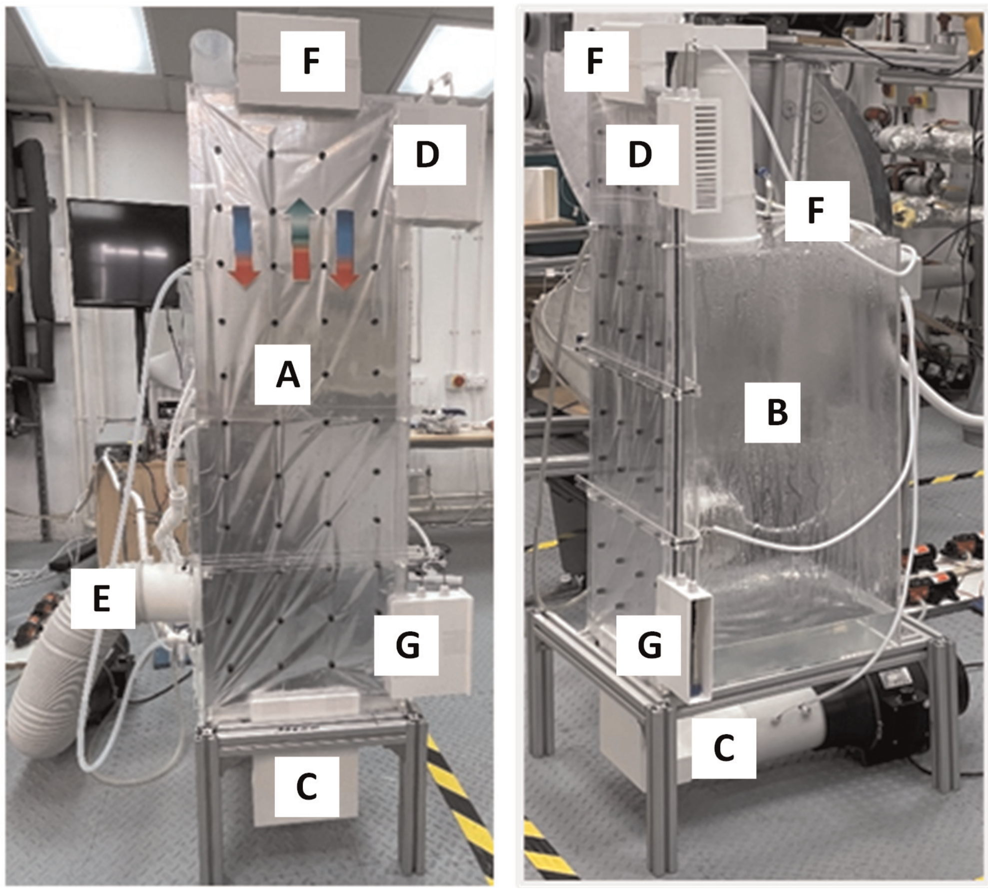

Figure 3.
Experimental test rig front and side view.

Parameter	Value
Heat exchanger length, mm	1000
Heat exchanger width, mm	300
Heat exchanger wall thickness, mm	0.025
Channel gap, mm	5
Mixing chamber height, mm	800
Mixing chamber width, mm	400
Mixing chamber length, mm	400

Table 1.
Geometric characteristics of the system.

inlet and outlet manifolds of the heat exchanger were developed using an in-house 3D printing facility. The design characteristics of the system are summarized in **Table 1**.

A real-time data acquisition system is used to monitor and record temperature data during experimentation. For this purpose, dry and wet bulb temperature sensors are installed at different locations in the system. General purpose thermistor probes are used for temperature measurements. For wet bulb temperature, the measuring station is developed using high capillary action felt material with a continuous water supply. Meanwhile, the flow rates in the system are measured using a hot wire anemometer (Testo 405i) at product air and working air outlets.

Parameter	Value
Outer air temperature, °C	29–43
Outdoor air humidity, g/kg	10
Outer air velocity, m/s	2.5
Working air velocity, m/s	2.5
Working air humidity, g/kg	18

Table 2.
Process parameters.

Parameter	Description	Accuracy
Temperature	General purpose thermistor probes (by OMEGA)	±0.15°C
Velocity	Hot wire anemometer (by Testo 405i)	±0.1 m/s
Humidity	Wet bulb measuring station (by OMEGA and customized)	±0.15°C
Data logging	Agilent Benchlink 34790a	

Table 3.
Instrumentation details.

Detailed experimentation of the developed system was conducted to capture the performance picture of the system. For this purpose, the system is operated at varying outdoor air temperature conditions. The process parameters considered in the study are presented in **Table 2**. The instrumentation details are summarized in **Table 3**.

2.3 Performance assessment

The performance of the cooler is measured in terms of heat extraction (cooling) capacity from the product air stream as it moves along the heat exchanger from 1 to 2. It is calculated in terms of airflow rate, specific heat, and temperature drop [40, 41].

$$\dot{Q}_{1-2} = \dot{m}_{PA}\, c_p\, (T_1 - T_2) \tag{1}$$

The heat rejected by the dry channel air stream during processes 1–2 is taken by the working air stream in the wet channel (3–4) as a combined latent and sensible heat. This heat transfer process is given as:

$$\dot{Q}_{1-2} = \dot{Q}_{3-4} = \dot{m}_{WA}\, c_p\, (T_4 - T_3) + \dot{m}_{vapor}\, \lambda_{fg} \tag{2}$$

The cooler performance is also measured in terms of the cooler performance indicator (CPI) which is the ratio of cooling produced to the energy consumed [42].

$$CPI = \frac{\dot{Q}_{1-2}}{\dot{E}_{input}} \tag{3}$$

The energy consumption is calculated in terms of fan energy and pumps energy used to maintain the required air and water flow rates, respectively. It can be used in terms of fluid flow rate, pressure differential, and component efficiency [43].

$$\dot{E}_{in} = \dot{E}_{blower} + \dot{E}_{pump} = \frac{\dot{V}_{air}\,\Delta P_{HE}}{\eta_{blower}} + \frac{\dot{V}_{water}\,\Delta P_{water}}{\eta_{pump}} \tag{4}$$

The pressure drop in the heat exchanger channels is calculated in terms of friction factor, channel length, flow velocity, and equivalent diameter [40, 44].

$$\Delta P_{ch} = f\,\frac{L}{D_h}\,\frac{\rho V_{air}^2}{2} \tag{5}$$

The fan and pump power are calculated using pressure drops of air and water with corresponding flow rates. The maximum power input calculated is 4 W which is constant for all operating conditions.

The above equations are valid under the following standard assumptions.

- Steady-state performance
- Negligible heat leak to-and-from the system
- The heat rejected by the product air stream is absorbed by the working air stream
- Supplementary water is fed at a constant temperature equivalent to a wet bulb during longer operational times
- Pressure drop in connections and manifold is negligible

3. Results and discussion

The performance of the proposed indirect evaporative cooler was investigated in terms of temperature drop, cooling capacity, and cooling performance index. The most important parameter in indirect evaporative coolers is the supply air temperature. This is because all other performance indices are governed by the supply air temperature. For this purpose, the cooler was tested under different outdoor air conditions to record the supply air temperature trend for 4-to-5-hour continuous operation. **Figure 4** shows the typical product air temperature trends for an outdoor air temperature of $40 \pm 0.5°C$. It shows that the product air was obtained at a uniform temperature of $25 \pm 0.5°C$. It implied the steady cooler performance during the whole operational time producing the uniform cool product air. Therefore, the cooler is suitable for continuous longer operations without the development of any longitudinal heat conduction effect or heat storage in heat exchanger walls. The visual inspection during experimentation shows the mist evaporation on the walls which extracts heat from the wall thus keeping it at the same temperature. Meanwhile, it is also important to emphasize that the product air was supplied at constant absolute humidity of 10 g/kg because of sensible cooling.

The temperature trends for the working air stream are shown in **Figure 5**. It is observed that the hot outer air entered at $40 \pm 0.5°C$ in the mixing chamber. It was cooled to the wet bulb temperature of $23 \pm 0.5°C$ by mixing with water mist. During

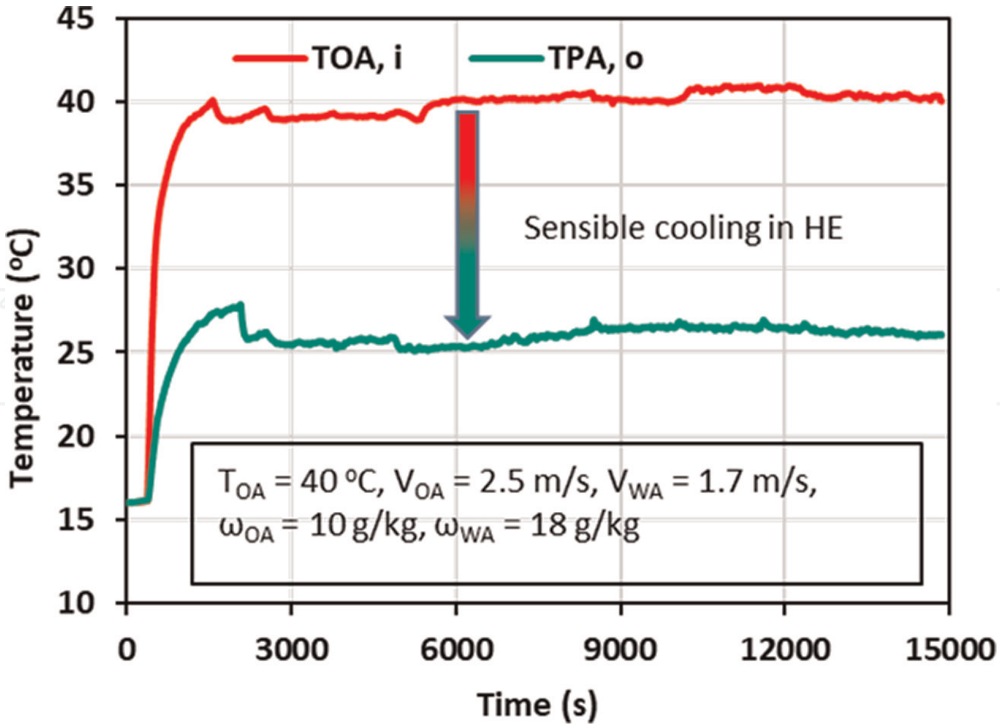

Figure 4.
Product air temperature trend.

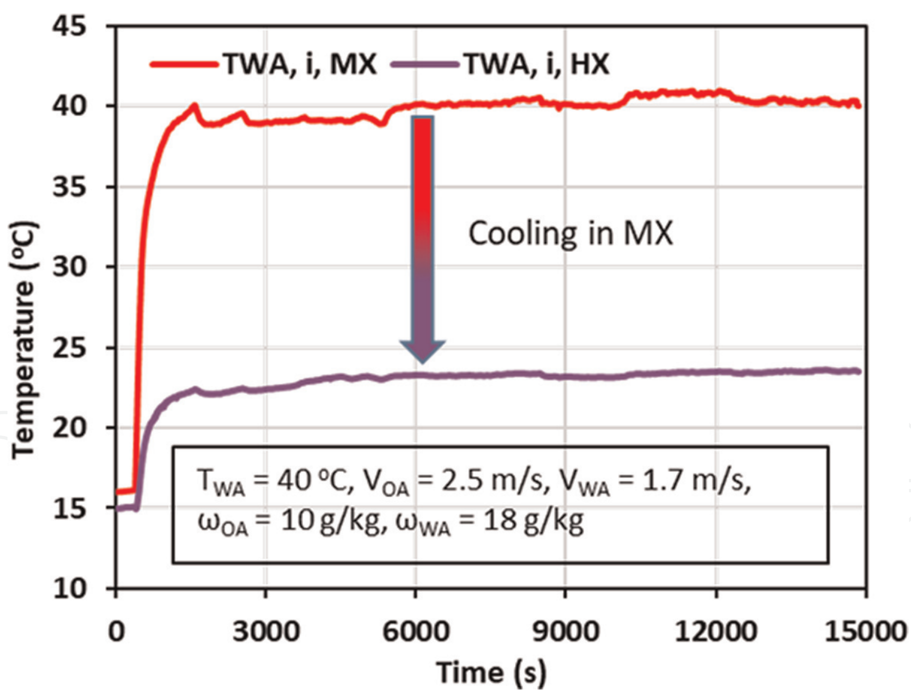

Figure 5.
Working air temperature trend.

mixing the working air achieved 100% relative humidity (ω = 18 g/kg). This cold and humid air then enters the heat exchanger and extracts heat from the product air stream. **Figure 6** shows the temperature trends for working air at the heat exchanger inlet and the product air at the heat exchanger outlet. It showed that the cooler performed close to wet bulb temperature with a maximum temperature differential across the air streams of 2–3°C thus giving the highest cooling performance.

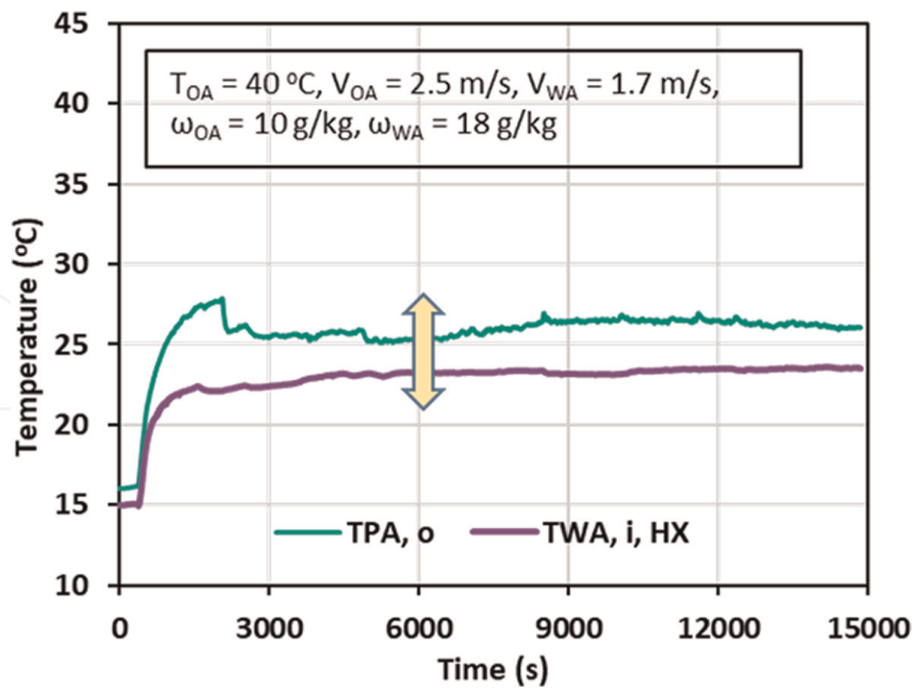

Figure 6.
Product air and working air temperature trend.

The performance of the cooler in terms of product air temperature at different outer air conditions is presented in **Figure 7**. It shows that the product air temperature increased as the outer air inlet temperature increased. A temperature rise of around 7° C (from 22.6 to 29.3°C) was observed for the outer temperature rising from 29 to 43° C. It suggested that the cooler can be used for any other higher or lower outer air temperature conditions with slight variation in the product air temperature. However, it is also important to emphasize that the variation in product air temperature is less

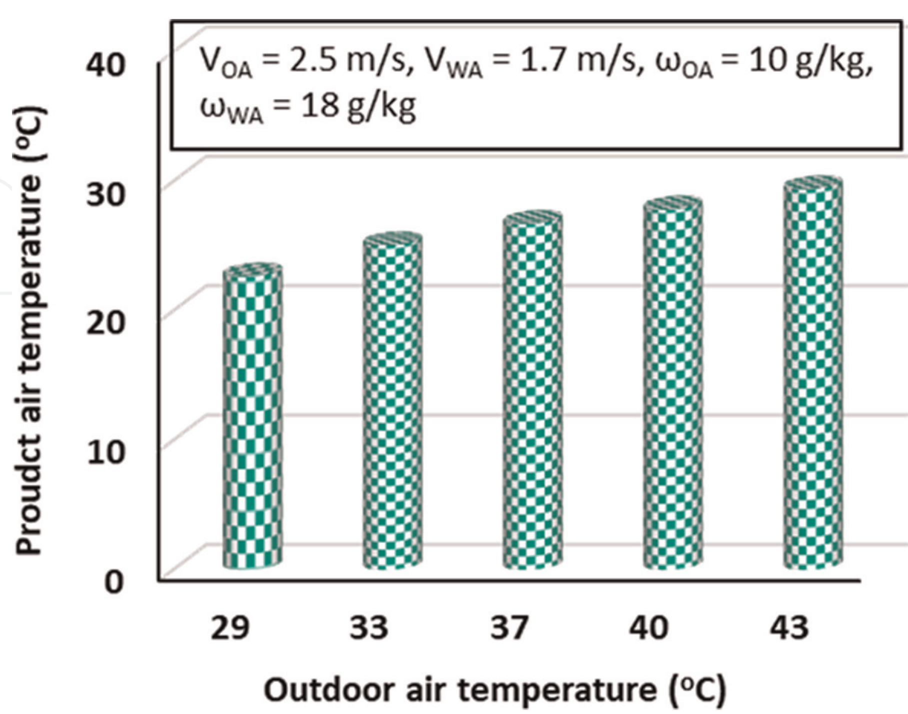

Figure 7.
Product air temperature trends at different outer air temperatures.

compared to the outer air conditions. It showed the performance of the cooler close to the wet bulb temperatures for all the operational conditions.

The effect of outer air temperature on the cooling capacity is presented in **Figure 7**. Correlated to the temperature drop, the cooling capacity of the cooler was observed to be increasing with the increasing outer air. For instance, the cooling capacity varied from 50 W to 110 W as the outer air temperature increased from 29 to 43°C. This increase in cooling capacity was achieved due to increasing temperature drop at higher outer temperature conditions. Similarly, the cooling performance index at different outer air temperature conditions is presented in **Figure 8**. It showed that the cooling performance index increased at higher outer air temperature conditions. The increase in CPI was due to an increase in the cooling capacity. For instance, the CPI increased from 13 to 28 as the outer air temperature increased from 29 to 43°C. Meanwhile, it is also worth mentioning that the power input was considered constant for all the cases because of the fixed pump and fan installation. However, less power is required (theoretically) at lower outer air conditions because of low water requirement and less working air velocity. Therefore, intelligent system control regulating the air and water supply commensurate to the outer air temperatures can offer the same higher CPI at lower outer air temperature conditions (**Figure 9**).

Besides promising energy performance, the proposed system also resolves the issue of high humidity in conventional water-based cooling systems. The outlet of the system for all operating conditions remains within the comfortable zones recommended by ASHRAE (Winter: RH = 75%, T = 21°C, Summer: RH = 53%, T = 27°C) and ISO (Winter: RH = 30–70%, T = 23°C, Summer: RH = 30–70%, T = 26°C). So, the issues associated with high humidity are eliminated while using the proposed system. One of the issues in this regard is Legionnaires which commonly occur due to airborne water droplets. It is important to emphasize that the supply air in the proposed system does not interact with water during any operational stage. Therefore,

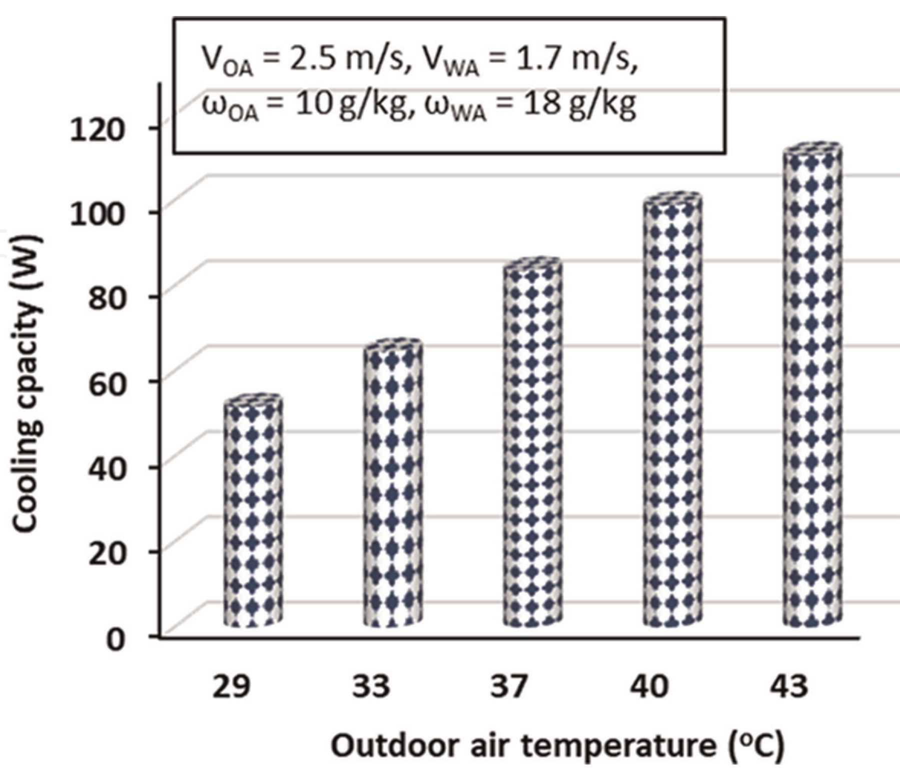

Figure 8.
Cooling capacity at different outer air temperatures.

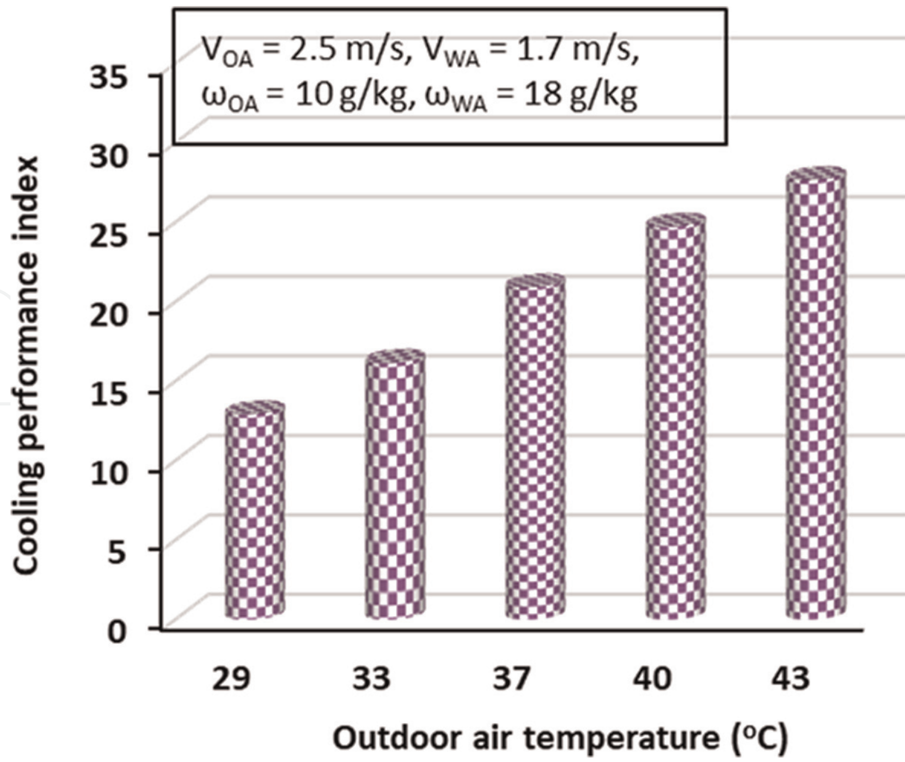

Figure 9.
Cooling performance index at different outer air temperatures.

there is no chance of mixing water droplets with the supply air stream and associated problems. Moreover, the working air can be used for beneficial use like hydroponics or water harvesting to further mitigate the potential issues with wet air delivery to the ambient.

The system is also convenient to integrate with the building particularly using the existing ducting network. This is because the system provides cool dry air at the required temperature as a single outlet. The outlet of the system can be connected to the ducting network in the buildings to supply air at different points. The duct fans can be used to facilitate air delivery at the required points by overcoming the pressure drop in ducts. It is also important to mention that in industrial-scale systems with high-velocity air delivery, the velocity of air is managed in the distribution duct to achieve the required comfortable range of supply air velocity as per ASHRAE standard 55.

4. Concluding remarks

An innovative indirect evaporative cooler is developed and tested experimentally. The proposed design is aimed to resolve the critical issues from manufacturing and operational viewpoints. The salient features of the proposed system are simple construction, low manufacturing, less maintenance, high operational reliability, and competitive cooling performance. The major outcomes of the study are summarized below.

- The system showed stable performance by cooling the product air to 25 ± 0.5°C from an outer air temperature of 45 ± 0.5°C.

- The product air temperature showed an increase of 7°C from 22.6 to 29.3°C as the outer air inlet temperature increased from 29 to 43°C. It suggested that the cooler can be used for any other higher of lower outer air temperature conditions with slight variation in the product air temperature.
- Correlated to the temperature drop, the cooling capacity of the cooler was observed to be increasing from 50 W to 110 W with increasing outer air from 29 to 43°C. This increase in cooling capacity was achieved due to increasing temperature drop at higher outer temperature conditions.
- The cooling performance index increased at higher outer air temperature conditions due to an increase in the cooling capacity. The CPI increased from 13 to 28 as the outer air temperature increased from 29 to 43°C.

Overall, the system showed promising performance with a maximum temperature drop of 14°C, a maximum cooling capacity of 110 W, and a maximum cooling performance index of 43°C. Meanwhile, the system supplied product air within the ASHRAE comfortable range for all the considered operational conditions. In addition to competitive performance, the system also addressed the limitations of the conventional systems through rearrangement and retrofitting of the component. Therefore, the proposed system can be a lucrative option for industrial-scale expansion of indirect evaporative coolers.

Acknowledgements

The authors would like to thank Northumbria University UK for funding under reference #RDF20/EE/MCE/SHAHZAD, and Northern Accelerator Proof-of-Concept award for AD4DCs (NACCF-232) Awarded to Dr. Muhammad Wakil Shahzad. Also acknowledged is the support provided by KAUST cooling initiative.

Conflict of interest

The authors declare no known conflict of interest.

Nomenclature

A	heat transfer area, m^2
c_p	specific heat
$\dot{E}$	energy, W
f	friction factor
$\dot{m}$	mass flow rate, kg/s
P	pressure, pa
$\dot{V}$	volume flow rate

Greek Symbols

Δ	change in quantity
λ	enthalpy

Subscripts

i	inlet
o	outlet
OA	outer air
PA	product air
WA	working air

Abbreviations

CPI	cooling performance index
IEC	indirect evaporative cooler

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