Experimental and parametric sensitivity analysis of a novel indirect evaporative cooler for greener cooling

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1. Introduction

The increasing global temperature (1.5–1.8 °C/century), population growth, improved living standards, and high urbanization rate has resulted in a rapid surge in cooling demand [1,2]. Particularly, in developing countries and densely populated areas, the temperature rise has been recorded at over 2 °C since 2018 [3]. So, to maintain the required temperature a huge amount of energy, and cost are invested in the cooling sector [4]. For instance, Fig. 1 shows that the air conditioner stock will surpass 5 billion by 2050 from a mere 1.6 billion in 2016 (10 new ACs/s for the next 30 years) [5]. The associated cooling energy consumption and emissions are also predicted to cross 6000 TWh and 170 Gigatons by 2050, respectively. These high energy consumptions are because of the extensive use (>90% of their market) of energy-intensive vapor compression cooling systems [6]. These systems operate on a typical simple cooling coil-based operational scheme which limits their performance. It also hinders efficiency improvements because of rigid temperature (5–7 °C) and pressure limitations across the components [7]. Therefore, their COP hovers around 3–4 with no significant progress in the last 3 decades. Moreover, the use of high global warming potential chemicals (HFCs, CFCs, etc.) make these systems environmentally unsafe [8]. Meanwhile, plenty of alternative options have been proposed and tested over the years as shown in Fig. 2. These include absorption, adsorption, desiccant, and chilled beams have not seen a wide range of acceptability for several reasons. These include large size, corrosive nature of chemical usage, compatibility of working pairs, large piping infrastructure, etc [9].

Water-based cooling systems have proven to be a favorable solution to these problems because of their simple operation, low energy
consumption, and economics [10]. However, direct evaporative coolers (one of the conventional methods) have limited applications due to high humidity (100%) issues [11,12]. Hence, these can only be used to ventilate open spaces in hot dry climates. While, in humid areas, confined zones, and populated spaces these systems cannot achieve thermal comfort [13,14]. So indirect evaporative cooling (IEC), a relatively new concept has emerged as a promising option for humidity-controlled cooling. In these systems, the room supply air undergoes sensible cooling without any humidity variations [15]. While another air stream (working air) is humidified, cooled, and used to extract heat from the supply air [16,17]. These two distinct air streams flow in the adjacent channels separated by a water-resistant conducting wall. This arrangement produces cool dry air by harnessing the air–water evaporative cooling potential [18]. Therefore, these can be used for human thermal comfort under diverse operating scenarios. In addition, these can also be used for data center cooling, food, pharmaceutical, commodities storage, poultry, and livestock comfort [19]. Plenty of research has been conducted on developing, testing, and improving indirect evaporative cooling systems [20,21].

The different aspects covered hitherto, include design improvements, operating parameters, flow arrangements, material developments, robust control systems etc. [22-26]. For instance, protrusions on heat exchanger plates improved the wet bulb efficiency ($\eta_{wb}$) by 30% more than the flat plate system because of high heat transfer [27]. Another study [28] showed that introducing fins on the dry side boosted system performance leading $\eta_{wb}$ to as high as 120%. The baffle-assisted channels offered up to 35% higher cooling capacity than the conventional plain channels [29].

### Nomenclature

<table>
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<tr>
<th>Symbol</th>
<th>Description</th>
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<tr>
<td>$c_p$</td>
<td>specific heat, J/kg</td>
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<tr>
<td>$H$</td>
<td>channel height, mm</td>
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<tr>
<td>$L$</td>
<td>length, mm</td>
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<td>$P$</td>
<td>power, W</td>
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<td>$Q$</td>
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<td>$\bar{X}$</td>
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### Greek letter

<table>
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<th>Symbol</th>
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<tr>
<td>$\omega$</td>
<td>humidity, g/kg</td>
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<tr>
<td>$\Delta$</td>
<td>change in quantity</td>
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<tr>
<td>$\eta$</td>
<td>efficiency</td>
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### Subscripts

- $i$: inlet
- $o$: outlet
- OA: outdoor air
- SA: supply air
- wb: wet bulb

### Abbreviations

- AC: air conditioner
- AFR: air flow rate ratio
- CFC: chlorofluorocarbons
- COP: coefficient of performance
- HFC: hydrofluorocarbons
- IEC: indirect evaporative cooler
- LPM: litre per minute
- MVC: mechanical vapor compression
- NSC: normalized sensitivity coefficient
- RH: relative humidity
- TWh: terawatt hours
- VC: vapor compression

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**Fig. 1.** Global AC unit stock, associated energy consumption, and emission trends [5].
reported performance improvements through plate enhancement using different patterns. A study by Sun et al. [32], showed that nozzle type and spray time greatly impact the IEC performance. Among different experimentally tested options, the spiral nozzle with a maximum 12 s spray time with a 1-minute gap turned out as the best operating scenario. Gulizzoni et al. [33] showed that water distribution from the top with working air in counterflow gives maximum performance with $\eta_{wb}$ 78–84.4%. In a similar study, Antonellis et al. [34] reported the highest $\eta_{wb}$ of 82–84% for the water distribution from the top. Meanwhile, the significant effect of outdoor air humidity has also been reported on IEC performance, particularly under dry channel side condensation conditions [35]. It was reported that the highly humid inlet air decreased IEC efficiency and increased water consumption [36]. Meng et al. [37] also reported a drop in orthogonal heat transfer because of additional water layer resistance on the dry channel side due to condensation conditions. The solution proposed for these problems was to dehumidify the inlet air before entering the dry channel [38]. It improved cooling capacity and energy efficiency by a maximum of 135% [39].

Besides, improvements in IEC performance have also been achieved by enhancing the evaporation process in the wet channel. For this purpose, the ideas of porous material-based construction [40] and hydrophilicity augmentation of the wall were proposed and tested. The latter proved to be more effective because of uniform water distribution, prolonged water retention, and a high evaporation rate [41]. Antonellis et al. [42] studied IEC constructed using aluminum sheets enhanced with hydrophilic coating on the wet side and reported $\eta_{wb}$ between 40 and 50%. Similarly, Chen et al. [43] reported a COP of 9 for IEC constructed using hydrophilic-coated aluminum sheets. Antonellis and Liberati et al. [44,45] studied hydrophilic coated IEC with dimples and reported a temperature drop of 6–17 °C, and dew point effectiveness of 62%. Ali et al. [46] studied IEC with circular fins of aluminum enhanced with wetting material and reported the $\eta_{wb}$ and COP as 37% and 14, respectively. Lin et al. [47] studied IEC constructed using plastic sheets enhanced with wicking material and reported an exergy efficiency of 42%. Duan et al. [48] investigated a polymer film-enhanced IEC and reported the wet bulb efficiency as 96–107%. Khalid et al. [49] estimated the wet bulb and dewpoint efficiencies of IEC constructed using felt-enhanced aluminum sheets as 92–120% and 62–85%, respectively. Similarly, the wet bulb and dewpoint efficiencies for IEC with Kraft paper as hydrophilic materials were calculated as 104–120% and 67–87%, respectively [50]. Shahzad et al. [51] studied IEC enhanced with felt material and multipoint air injection. They reported that the multipoint air entry increased temperature drops by 2 to 3 °C thus achieving the COP cooling between 37 and 78 for different operating conditions. In the meantime, it is also important to emphasize that in IEC technology, M-cycle based systems have shown superior performance because of utilizing the cold dry air in the wet channel [52,53]. It improves the overall system performance because of low temperature in the wet channel [54,55]. One of the most comprehensive recent studies on M-cycle IEC concluded that M-IEC can be used for various applications like standalone cooler, energy recovery section for MVC systems power recycles cooling, heat recovery, water distillation/ desalination.

![Fig. 2. Different cooling options [9].](image-url)
The review of earlier studies suggests that IEC can be a viable technology for energy-efficient and humidity-controlled cooling with promising results from lab-scale investigations. Meanwhile, it is also important to note that in all the above systems the construction material needed hydrophilicity enhancement on the wet channel side for satisfactory performance. It was achieved by supplementing the wall with high water affinity materials like felt, wick, cotton, coating, etc. The special wall treatment makes commercial-scale development of the system (with hundreds of channels) complex and arduous. This is because the two surfaces are joined together using adhesive material (glue) that loses sticking power with time due to high temperature and humidity conditions. So due to heavy weight after soaking in water the two surfaces detach thus blocking the narrow air channels are blocked thus mitigating the cooling capacity. Therefore, the commercial-scale development and testing of these systems have been seldom reported in the literature. The current study proposed and investigated a novel indirect evaporative cooler constructed using high thermal conductivity plain aluminum sheets. An auxiliary humidifier is retrofitted with the indirect evaporative cooler constructed using high thermal conductivity plain aluminum sheets. An auxiliary humidifier is retrofitted with the heat exchanger to maximize the air–water humidification. An experimental generic cell based on the proposed arrangement was developed and tested. A vis-a-vis comparison and detailed working principle of the conventional and proposed systems are presented in the subsequent section.

2. The existing and proposed IEC configurations

The indirect evaporative cooler (IEC) configuration considered in the current study offers several benefits compared to the existing conventional systems through modified design. This is because almost all existing IECs operate on the same operational scheme where the outdoor air is cooled in the dry channel of the heat exchanger through the heat transfer to the working in the adjacent wet channel as shown in Fig. 3. In these systems, the hot outdoor air enters the dry channel (1) and rejects its heat to achieve supply air temperature (2) along the heat exchanger (1–2) to the wet channels. Where the hot outdoor air also enters the wet channel (3) and water is showered as it moves along the heat exchanger (3–4). This water showering reduces its temperature and increases the humidity which extracts sensible and latent heat from the dry channel stream. In these systems, the wet channels are designed so that the working air enters from one side and is humidified through continuous water showering inside the heat exchanger. This water showering reduces the temperature through humidification and evaporation of the air and the humidity of this air stream is increased (approach to RH = 100%). However, this configuration needs special treatment (i.e., high hydrophilicity) of the heat exchanger surface on the wet side to increase the water retention time for prolonged air–water contact. Meanwhile, it is important to note that the dry channel side is required to be hydrophobic to prevent any moisture transfer between the channels for humidity-controlled cooling which is the main purpose of IEC configuration. Therefore, the development of the heat exchanger wall becomes complex and requires careful monitoring during operation. This is because continuous water showering on the hydrophilic surface poses various issues. For instance, water distribution, accumulation, collection, heavyweight, sagging, detachment from the surface, and microbial growth are some of these major issues.

While in the proposed system the multi-surface hydrophilic-hydrophobic wall is replaced by a simple aluminum sheet thus removing the associated problems. However, an additional humidifier is retrofitted with the heat exchanger to achieve working air humidification and cooling. The system works as shown in Fig. 4 where the outdoor air at high temperature enters (1) the dry channel using an axial fan. While the working air first enters the humidifier (3) where it interacts with a fine mist created using a nozzle and pump. This air–water mixing in a humidifier cools down the high-temperature air to achieve wet bulb temperature and 100% relative humidity (4). This cold humid air then passes through the working air channels of the heat exchanger thus extracting heat from the dry channel (4–5). The orthogonal heat transfer between the channels results in the cooling of the dry air to the desired supply air temperature which is supplied to the cooling space (1–2). While on the working air side, the evaporation of carryover mist takes place resulting in a slight decrease in the humidity and increase in the temperature because of latent and sensible heat transfer, and this high humidity hot air is discarded to the ambient.

The outdoor air and working air handling process are presented on the psychrometric chart as shown in Fig. 5. It shows that the outdoor air undergoes sensible cooling from 1 to 2 where its temperature drops to supply air temperature. Where the working air first enters the humidifier where it is humidified and cooled from 3 to 4. During this process, the temperature of the working air is dropped to the wet bulb temperature, and the humidity is increased to 100%. This cold humid air then flows through the working air channel from 4 to 5 where it extracts heat from the dry channel.

![Fig. 3. The conventional indirect evaporative cooler schematic and air–water flow scheme.](image-url)
2.1. Experimental system

A generic cell based on the proposed IEC scheme was designed, fabricated, and tested experimentally as shown in Fig. 6. The system heat exchanger constructed using a simple aluminum sheet resolves the major design issues of the earlier systems as shown in Fig. 7. The system does not involve any hydrophilic material and all the maintenance-demanding parts like the pump, fan, nozzle, and humidifier are arranged outside the heat exchanger with easy access without opening the main cooler. Similarly, water–air interaction occurs in the humidifier where it can be optimized without changing the cooler configuration. Moreover, the wet channel wall has a high thermal conductivity which increases the orthogonal heat transfer rate eliminating the additional resistances due to the felt material and water layer. Furthermore, the complexities and costs associated with heat transfer surface development (e.g., sticking of wicking material, etc.) evade because of simple aluminum foil use.

The humidifier consists of an airtight container with a working air inlet, working air outlet, nozzle, supplementary water supply system, float valve, and water temperature sensor. The dimensions of the humidifier are selected as a length 800 mm and a diameter 400 mm to achieve proper air–water mixing. A water spray nozzle is installed at the top with 360° coverage close to the working air outlet. The hot outdoor air enters the humidifier from the bottom and the water is sprayed from the top. This counter-current flow enhances the humidification process, and the air achieves 100% relative humidity and low wet bulb temperature at the humidifier outlet. The water supplied is calculated based on the humidity difference at the inlet and outlet of the humidifier. However, in the current case additional 30% water is sprayed to achieve mist carryover in the wet channel for maximum cooling through evaporation in the channel. For this purpose, a 12 V DC pump with 3.8 LPM at a delivery pressure of 2.4 bar is used. The nozzle attached to the pump provides fine mist which enhances the atomization of water droplets and boosts evaporation. The pump inlet relates to the humidifier sump which recirculates water to minimize water consumption. Meanwhile, a supplementary water supply is connected through a float valve to the humidifier to maintain the water level in the humidifier sump. The humidifier performance can be further enhanced by adding advanced packing materials that can help to reduce the size and area footprints of the system. The geometric and operational parameters used in the current study are presented in Table 1.

The system developed is tested at different outdoor air temperature conditions. The system is fully instrumented to monitor and record the temperature data at all inlets and outlets. For this purpose, the dry and wet bulb temperatures are measured at the dry channel inlet, wet channel inlet, dry channel outlet, humidifier inlet, and humidifier outlet. Omega general-purpose thermistor probes are used for recording the temperature and a hot wire anemometer is used for velocity measurements. Agilent bench-link data logger is used for data acquisition from sensors.

3. Governing equations

The generic cell performance was measured in terms of different output parameters frequently used for IEC performance in earlier studies. These parameters include temperature drop (ΔT), cooling capacity (Q), coefficient of performance (COP), and wet bulb efficiency (ηwb). The mathematical formulation for these parameters is given below.

Various temperature differentials are calculated in the IEC system for instance working air inlet and outlet temperature difference across the heat exchanger, dry and working air temperature differences at the inlet and outlet of the heat exchanger, temperature drop in working air across the humidifier, and the log mean temperature difference across the heat exchanger. However, the most significant parameter for indirect evaporative cooler performance measurement is the temperature drop achieved across the dry channel (ΔT0A). This is because the outlet of the dry channel is supplied in the air-conditioned space for cooling. This
temperature drop is further used to estimate the cooler performance and efficiency of the cooler. It is calculated as the difference between hot outdoor air inlet temperature and cold supply air outlet temperature as given below.

\[ \Delta T_{OA} = T_{OA,i} - T_{SA,o} \] (1)

The cooling capacity indicates the cooling/heat extraction rate from the primary air stream and is calculated using temperature drop and mass flow rate as.

\[ \dot{Q} = \dot{m}_{OA} \cdot c_p \cdot \Delta T_{OA} \] (2)

The coefficient of performance (COP) represents the ratio of cooling produced and the power invested to achieve that cooling and is calculated as.

\[ COP = \frac{\dot{Q}}{P_{total}} \] (3)

2.2. Sensitivity analysis

Sensitivity analysis is an important procedure to categorize the significance of system parameters to determine the most critical for design and performance [56]. It helps to control the system performance systematically and also highlights the roadmap for subsequent parametric optimization [57]. In this regard, calculus-based sensitivity analysis is among the most frequently used methods because of its high outcome reliability and robust analysis. It works by modeling each independent parameter in the system as a sum of its nominal value (\(X\)) and the perturbation (\(U\)) as [58].

\[ X = \bar{X} \pm \Delta X \] (5)

Likewise, the perturbation in the output parameter representing the system performance is estimated in the differential form as [59],

\[ \frac{\Delta Y}{\Delta X} = \frac{dY}{dX} \] (6)

Meanwhile, it is important to emphasize that in thermal systems like IEC, each performance parameter is governed by several input parameters and hence cannot be individually modeled. In such cases with the multivariate function \( Y = Y(X_1, X_2, \ldots, X_n) \), the sensitivity is modeled in terms of probations of all relevant input parameters which is finally calculated as the root sum square product of the individual perturbation computed to the first-order accuracy, as given below [60].
where in the above equation $U_Y$ and $U_X$ are the perturbations in $Y$ and $X$, while, $Y$ and $X$ are their nominal values \[66\].

For each performance parameter, the above general equation can be modified in terms of independent parameters. The corresponding sensitivity measurement equations for Cooling capacity, COP, and $\eta_{wb}$ are given as.

\[
\frac{U_Q}{Q} = \left[ \frac{\left( \frac{\partial Q}{\partial T_p} \right)^2}{Q} \left( \frac{U_{T_p}}{Q} \right)^2 + \frac{\left( \frac{\partial Q}{\partial T_{in}} \right)^2}{Q} \left( \frac{U_{T_{in}}}{Q} \right)^2 + \frac{\left( \frac{\partial Q}{\partial T_{wb}} \right)^2}{Q} \left( \frac{U_{T_{wb}}}{Q} \right)^2 \right]^{1/2}
\]

\[
\frac{U_{COP}}{COP} = \left[ \frac{\left( \frac{\partial COP}{\partial T_p} \right)^2}{COP} \left( \frac{U_{T_p}}{COP} \right)^2 + \frac{\left( \frac{\partial COP}{\partial T_{in}} \right)^2}{COP} \left( \frac{U_{T_{in}}}{COP} \right)^2 + \frac{\left( \frac{\partial COP}{\partial T_{wb}} \right)^2}{COP} \left( \frac{U_{T_{wb}}}{COP} \right)^2 \right]^{1/2}
\]

\[
\frac{U_{\eta_{wb}}}{\eta_{wb}} = \left[ \frac{\left( \frac{\partial \eta_{wb}}{\partial T_p} \right)^2}{\eta_{wb}} \left( \frac{U_{T_p}}{\eta_{wb}} \right)^2 + \frac{\left( \frac{\partial \eta_{wb}}{\partial T_{in}} \right)^2}{\eta_{wb}} \left( \frac{U_{T_{in}}}{\eta_{wb}} \right)^2 + \frac{\left( \frac{\partial \eta_{wb}}{\partial T_{wb}} \right)^2}{\eta_{wb}} \left( \frac{U_{T_{wb}}}{\eta_{wb}} \right)^2 \right]^{1/2}
\]

2.3. Error propagation analysis

The error propagation analysis was conducted for performance parameters against input parameters using the approach presented earlier for IECs \[41\]. For this purpose, the uncertainty in the measurement of temperature is considered as $\pm0.15^\circ C$ (the accuracy of the sensors). Similarly, the uncertainty for velocity measurement is taken as $\pm0.5$ m/s.
It is observed that the total error in the calculated parameters i.e., $Q$ is $\pm 0.0167$, COP is $\pm 0.527$, $\eta_{wb}$ is $\pm 0.867$. It shows that the values presented for these parameters are reasonably accurate compared with the earlier studies.

4. Results and discussion

4.1. Experimental testing

The performance of the generic cell was assessed experimentally at different outdoor air temperatures varying from 28 to 43 °C. Fig. 8 presents the effect of outdoor air temperature on the supply air temperature, and the temperature drop trend. It was observed that the supply air temperature increased 22 to 24 °C for the outdoor air temperature varying from 28 to 43 °C. It showed a slight increase in the supply air temperature with increasing outdoor air temperature. However, the maximum cooling effect in terms of temperature drop was obtained at the peak outdoor air temperature of 18 °C which dropped to 5.5 °C for the outdoor air temperature of 28 °C.

Fig. 9 presents the cooling capacity trend against outdoor air temperature. It showed that the maximum cooling capacity of 100 W was obtained at 43 °C which dropped to 31 W, as the outdoor air temperature decreased to 28 °C. This drop in cooling capacity is due to decreasing temperature gradient between dry and wet air streams.

Similarly, the effect of outdoor air temperature on the coefficient of performance of the system is presented in Fig. 10. It showed that the maximum COP of 31.27 was also obtained at the maximum outdoor air temperature of 43 °C. While it decreased to 9.5 for the outdoor air temperature of 28 °C. Meanwhile, it is important to emphasize that the COP presented here is only for cooling and does not involve any dehumidification process which may be required for high-humidity outdoor air conditions. Moreover, the COP is calculated for the maximum constant input power of 3.2 W required for fans and pump. This power input is calculated based on air velocity (1.7 m.s$^{-1}$) in channels and water requirements. The low air velocity results in a very low-pressure drop in the channels thus needing low fan power. Similarly, low water consumption needs a small pump for air water mixing in the humidifier. So, even after the integration of low performance (COP$_{deh}$ 1–1.5) dehumidification section the overall system COP is expected to be around 10–12. Furthermore, the proposed system does not involve any energy-intensive compressor and simply needs low-energy consumption fans and a pump for air–water-based cooling. Therefore, it achieves considerably (2–3-fold) high COP than the vapor compression (VC) cooling systems (COP$_{VC}$ 3–4) which use a compressor to maintain rigid thermal and pressure lifts across the evaporator and condenser.

Similarly, the effect of outdoor air temperature on the wet bulb efficiency is presented in Fig. 11. The maximum efficiency was calculated as 93% at an outdoor air temperature of 43 °C which dropped to 65% at an outdoor air temperature of 28 °C. Overall, the study inferred that the maximum cooler performance was achieved at the highest outdoor air temperatures thus making it applicable to harsh climatic conditions.

Besides outdoor air temperature, the effect of outdoor air velocity, air flow rate ratio, and the water temperature has also been investigated for the system. The preliminary investigation in this regard shows that as the velocity of outdoor air is reduced, the supply air temperature slightly reduces. However, the cooling capacity of the system also reduces which indicates that the system underperforms the designed capacity. Similarly, a rise in velocity also increases the supply air temperature which exceeds the ASHRAE comfortable zone requirements thus making the system unsuitable for human thermal comfort. Therefore, the velocity should be managed accordingly to achieve maximum cooling potential. Similarly, a decrease in the working air velocity increases the supply air temperature and an increase in working air velocity increases the input power. The system is further investigated for optimal outdoor air velocity, air flow rate ratio, and water parameters.

4.2. Sensitivity analysis

The parametric sensitivity analysis of important input parameters including outdoor-and-working air temperature and velocity on the system performance was conducted. The performance indicators included cooling capacity, coefficient of performance, and wet bulb efficiency. The sensitivity of input parameters is presented in terms of normalized sensitivity coefficients (NSCs) as shown in Fig. 12. It was observed (refer to Fig. 12 (a)) that the sensitivity of the input parameter...
for cooling capacity follows the order as $T_{OA,i} > T_{SA,o} > V_{OA} > T_{WA,o} > T_{WA,i} > V_{WA}$ with sensitivity coefficients as 3.9, 1.4, 1.0, 0.7, 0.3 and 0.2, respectively. While for the coefficient of performance (refer to Fig. 12 (b)) the sensitivity of parameters follows the order as $T_{OA,i} > V_{OA} > T_{WA,o}$ with NSCs as 3.9, 1.0, and 0.7, respectively. The other parameters are almost insensitive to the coefficient of performance. Similarly, for wet bulb efficiency (refer Fig. 12 (c)) the sensitivity coefficients follow the order as $T_{SA,i} > T_{WA,i} > T_{OA,i}$ with NSCs as 19.2, 16.3, and 0.9, respectively. The other parameters do not influence the wet bulb efficiency.

Besides the point-based sensitivity of input parameters, the variation in the sensitivity of two important parameters i.e., $T_{OA,i}$ and $T_{SA,o}$ was also studied over the operating range. Fig. 13 presents the effect of outdoor air temperature on NSCs of the coefficient of performance and wet bulb efficiency keeping other variables constant. The analysis showed that the NSCs of COP and $\eta_{wb}$ decreased as $T_{OA,i}$ increased. This is because of the collective effect of all other constant parameters and perturbations induced. However, it is also important to mention that the parametric values of COP and $\eta_{wb}$ increased with increasing outdoor air temperature (refer to section 4.1). Similarly, Fig. 14 shows the effect of supply air outlet temperature on NSCs of the coefficient of performance.
and wet bulb efficiency for the constant value of other parameters. It was observed that the NSCs of COP and \( \eta_{wb} \) increased with increasing supply air temperature because of combinatory impacts of constant quantities and perturbation.

5. Performance comparison and future roadmap

The performance of the current system is compared with the traditional experimental systems in terms of maximum temperature drop and the maximum coefficient of performance. For this purpose, three different systems which have been reported to achieve commendable cooling performance are selected. The first system consists of a regenerative IEC system enhanced with felt material developed and tested by Jie et al. [67]. They reported that regenerative arrangement improves the cooling performance of the system. The second system consists of a tubular ceramic IEC system presented by Wang et al. [40] and Sun et al. [32]. The ceramic material improved the water evaporation rate thus decreasing the pump work and improving the overall system performance. The third system was developed by Comino et al. [68] which consists of a regenerative indirect evaporative cooler where the cold dry air was purged in the working air channel to improve system performance. A comparison of these systems with the current system along with geometric and operating parameters is presented in Table 2. It shows that the maximum temperature drop achieved by the current system is 18.16 °C which is competitive with the existing system with 15, 17, and 19 °C. Similarly, the COP of the current system is 31.3 which is competitive with the traditional systems with 21, 30, and 32.

The study shows that the proposed system achieves competitive performance in terms of temperature drop and coefficient of performance compared to the traditional systems. In addition, the proposed scheme also addresses the major design limitations in traditional systems and has a significant potential to outperform conventional vapor compression systems for cooling. The future work direction for the proposed system is the development and testing of a commercial large-scale unit. A rigorous experimental investigation encompassing the effect of all geometric and process parameters testing is required before the deployment of the system on actual working sites. Moreover, the system also needs to be evaluated as an energy recovery unit for MVC systems. This is because IEC can take a significant (50–60%) of the sensible cooling load of MVC systems which can reduce energy consumption.

6. Conclusion

Experimental and normalized sensitivity analysis of a novel indirect evaporative cooler is presented in this study. The system is developed based on an advanced operational cycle with the improved arrangement of components to achieve high heat transfer, better operational reliability, low maintenance vulnerability, and simple and economical manufacturing compared to conventional systems. The system was tested under different outdoor air temperature conditions. The investigations involved experimental study followed by numerical parametric sensitivity analysis. The major findings of the study under considered operation conditions are summarized below.

- The supply air temperature for the cooler varied between 22.46 and 24.86 °C for the outdoor air temperature varying 28 to 43 °C. This shows that the cooler operates close to wet bulb temperature and hence the supply air temperature is marginally increased by 3 °C against a large increase of 15 °C in the outdoor air temperature.
- The cooler achieved a maximum cooling capacity of 100 W at a maximum outdoor air temperature of 43 °C. It shows superior cooler performance at higher outdoor air temperatures which makes it suitable for harsh climatic conditions.
- The maximum coefficient of performance of the system was calculated as 31 at 43 °C outdoor air temperature which dropped to 9 at the outdoor air temperature of 28 °C.
- The wet bulb efficiency values were calculated as 93% and 65% for an outdoor air temperature of 43 °C and 28 °C, respectively. It also confirms the superior cooler performance at higher temperatures.
- The parametric sensitivity analysis reported the most sensitive parameters for cooling capacity as \( T_{OA,i} > T_{WA,i} > V_{OA} > T_{WA,o} > T_{WA,j} > V_{WA} \). While, for the coefficient of performance as \( T_{OA,i} > V_{OA} > T_{WA,o} \) and the wet bulb efficiency as \( T_{WA,i} > T_{WA,j} > T_{OA,i} \). All other parameters were observed to be insignificant to the cooler performance.
Fig. 12. Sensitivity coefficients of (a) cooling capacity, (b) coefficient of performance, and (c) wet bulb efficiency.
Overall, the system showed competitive performance with the conventional indirect evaporative cooling systems by achieving a maximum temperature drop of 18 °C, coefficient of performance as 31, and wet bulb efficiency as 93%. In addition, the improved configuration also holds several advantages like high operational life, low maintenance, low cost, and resilient design. The proposed system can be expanded on a commercial scale because of promising cooling performance and robust design to realize the greener cooling goals for sustainable development. The future work for the proposed system involves development at a commercial scale and detailed experimental testing under assorted operating conditions.

Fig. 13. Sensitivity coefficients of coefficient of performance and wet bulb efficiency against outdoor air temperature.

Fig. 14. Sensitivity coefficients of coefficient of performance, and wet bulb efficiency against supply air temperature.
Table 2
Performance comparison for proposed and traditional systems.

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<th>System description</th>
<th>Parameters</th>
<th>COPmax</th>
<th>Ref.</th>
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<tr>
<td>Current</td>
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<tr>
<td>Humidifier-assisted counterflow IEC</td>
<td>L = 1100 mm, W = 300 mm, HTR = 5 mm</td>
<td>18.16</td>
<td>31.3</td>
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<tr>
<td></td>
<td>θOA = 28–43 °C, T0,a = –10–11 g/kg, VOA = 1.7 m/s, AFR = 100%</td>
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</tr>
<tr>
<td>Regenerative IEC</td>
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</tr>
<tr>
<td>Polyethylene Terephthalate (PET) sheets with felt material</td>
<td>L = 600 mm, W = 150 mm, H4,5 = 3 mm</td>
<td>19</td>
<td>30 [67]</td>
</tr>
<tr>
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<td>θOA = 30–40 °C, T0,a = 10.5–12.0 g/kg, VOA = 1.1–2 m/s, AFR = 0.2–0.8</td>
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<td>Tubular IEC</td>
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<tr>
<td>Ceramic porous tubes</td>
<td>L = 600 mm, Dtube = 30</td>
<td>15</td>
<td>32 [32,40]</td>
</tr>
<tr>
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<td>θOA = 34–45 °C, RH4,5 = 31–53%, mpu = 200–500 m³/h, mnu = 350 m³/h, mwa = 1.5 m³/h</td>
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<td>Regenerative IEC</td>
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<tr>
<td>L = 45 mm, W = 30 mm, H = 3 mm</td>
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<td>21</td>
<td>[68]</td>
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<td>mpu = 1530 m³/h, AFR = 0.44, T4,5 = 37.78 °C, T0,a = 9.05 g/kg</td>
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</table>

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

No data was used for the research described in the article.

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References
