

# **A First-of-Its-Kind Two-Stage Dew-Point Evaporative Cooler with High Energy Efficiency and Compact Design**

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## **Abstract**

This research addresses the long-standing high working air ratio problem of conventional single-stage dew-point evaporative coolers (DPECs) by developing a first-of-its-kind two-stage DPEC. The system introduces a novel indirect cooling mechanism that distributes the cooling load across two sequential stages, significantly reducing the product air required for evaporation and enhancing energy efficiency. The key component of the cooler is a compact heat and mass exchanger capable of handling four fluid flows within a single unit, ensuring high performance and space efficiency. Tested under subtropical climate conditions, the developed two-stage DPEC demonstrated exceptional performance, preserving 88–90% of the product air while maintaining its temperature below the wet-bulb and near the dew point. Additionally, it achieved a coefficient of performance (COP) of 15–17 under actual operating conditions, far surpassing conventional air conditioning systems. These results highlight its superior cooling capacity, energy efficiency, and compactness compared to existing single-stage M-cycle DPECs. The two-stage DPEC, with its high efficiency and compact design, is a scalable and sustainable cooling solution for real-world heating, ventilation, and air

conditioning (HVAC) systems, offering significant energy savings and improved thermal comfort. This study underscores its potential as a transformative technology for reducing energy consumption and addressing cooling demands in subtropical and tropical climates.

**Keywords:** Two-stage evaporative cooling, dew-point cooler, heat and mass exchanger, sustainable cooling solution

## Nomenclature

|           |                            |
|-----------|----------------------------|
| $A$       | Heat exchanging area       |
| $E$       | Fan power                  |
| $e$       | Uncertainty of measurement |
| $f$       | Correlation function       |
| $\dot{m}$ | Mass flowrate              |
| $n$       | Number of variables        |
| $Q$       | Cooling capacity           |
| $q$       | Mean heat flux             |
| $r$       | Product air ratio          |
| $T$       | Air temperature            |
| $\dot{V}$ | Air volume flowrate        |
| $x$       | Measured variable          |

### *Greek symbols*

|        |                        |
|--------|------------------------|
| $\eta$ | Efficiency of used fan |
| $\xi$  | Cooling effectiveness  |

### *Subscripts*

|       |          |
|-------|----------|
| $dp$  | Dewpoint |
| $in$  | Input    |
| $pro$ | Product  |
| $wb$  | Wet-bulb |

### *Abbreviations*

|      |                                        |
|------|----------------------------------------|
| AC   | Air Conditioning                       |
| COP  | Coefficient of performance             |
| DPEC | Dew-point evaporative cooler           |
| HR   | Humidity ratio                         |
| HVAC | Heat, Ventilation and Air Conditioning |
| RH   | Relative humidity                      |

## 1. Introduction

Air conditioning is essential in our modern society [1]. It plays a vital role in improving human productivity and well-being, especially in hot regions [2–4]. Additionally, air conditioning is crucial for preserving temperature-sensitive goods and equipment, such as pharmaceuticals, food products, and data servers, ensuring their safe storage and reliable operation [5]. However, air conditioning is primarily based on mechanical vapor-compression cycles. This has been causing significant environmental impact, raising numerous ecological concerns. The fluorinated refrigerants used in air conditioning systems have high potential for ozone-depletion and global warming. Furthermore, the indirect emission from the electricity consumption, which is required to operate air conditioning systems, contributes to their carbon footprint, particularly in regions where fossil fuels dominate power generation [6]. It is reported that the widespread use of air conditioning systems imposes substantial demands on global energy resources. HVAC systems account for nearly half of the energy consumption in the building sector in developed nations. In tropical and subtropical regions, the share of energy dedicated to air conditioning is even higher [7]. Besides, air conditioners pose health and ventilation challenges, especially in poorly designed systems. This is because inadequate ventilation can lead to indoor pollutants, respiratory issues, and reduced air quality [8]. The COVID-19 pandemic underscored the importance of proper ventilation in air-conditioned environments [9].

Balancing thermal comfort, energy efficiency, and ventilation requirements remains a critical challenge in air conditioning system design and operation. Evaporative coolers offer an energy-saving alternative to traditional air conditioning systems. These cooling devices utilize the latent heat of water evaporation to achieve temperature reduction [10]. Compared to conventional air conditioning (AC) systems, this technology presents several notable advantages. The primary benefit lies in its higher energy efficiency. Additionally, both the initial investment and operational costs are relatively lower. The installation and maintenance processes are also simpler. From an environmental perspective, this technology does not reject harmful chemicals or heat to the atmosphere [11]. Indirect evaporative coolers represent a specialized class of evaporative cooling technology that separates air streams into distinct wet and dry channels. This design facilitates heat exchange while preventing moisture transfer, making indirect evaporative coolers ideal for applications where maintaining low humidity levels is essential. However, the performance of these systems is constrained by the wet-bulb temperature of the process air, which limits the resultant air temperature to this critical threshold. Consequently, indirect evaporative cooling systems are categorized under wet-bulb

1    evaporative cooling technology [12].

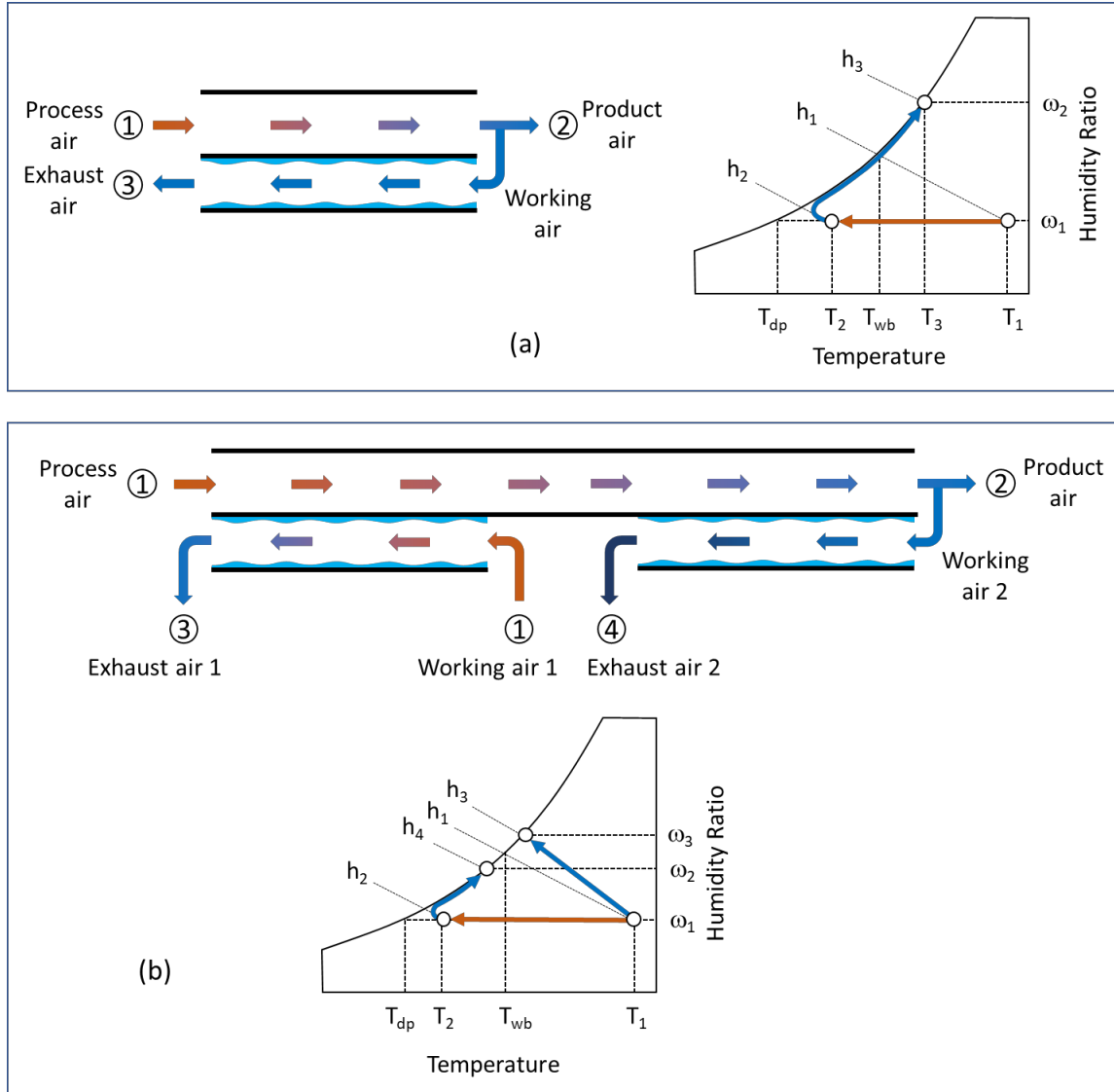


**Figure 1:** (a) conventional single-stage M-cycle dew-point cooling, and (b) proposed two-stage indirect dew-point evaporative cooling

The evolution of technology from traditional wet-bulb indirect evaporative coolers to advanced dew-point indirect evaporative coolers (DPECs) has been significantly advanced by the pioneering research of Dr. Maisotsenko, as illustrated in Figure 1(a). This innovative system, commonly known as the M-cycle cooling system, effectively integrates heat transfer and water evaporation to achieve superior cooling performance [13,14]. In an M-cycle cooler, a portion of the cooled air from the dry channel is redirected into the wet channel, where it serves as the working air for the DPECs. This pre-cooled working air enhances the overall cooling capacity, enabling performance that surpasses that of conventional indirect evaporative coolers. As a result, this cooling system can produce air temperatures that not only drop below

the wet-bulb temperature but also approach the dew-point temperature of the process air. This performance is further illustrated in the psychrometric chart shown in Figure 1(a) [15–18].

Recent research on DPECs has focused on enhancing cooling performance and energy efficiency through both simulation and experimental methodologies. Simulation analyses have been widely employed to evaluate thermodynamic behavior and optimize system parameters, offering acceptable accuracy at low cost and within short development cycles. These models have progressed from one-dimensional [19,20] and two-dimensional [21–27] configurations to sophisticated three-dimensional simulations [28–30]. On the experimental front, researchers such as Duan *et al.* [31], Xu *et al.* [32], Jia *et al.* [33], Riangvilaikul and Kumar [34], Khalid *et al.* [35], Lin *et al.* [36,37], and Wang *et al.* [38,39] have designed and tested full-scale DPEC prototypes under various ambient conditions, effectively validating their real-world performance. In addition, studies by Liu *et al.* [40] and Iyer *et al.* [41] have successfully combined simulation and experimentation to design and evaluate high-performance systems, confirming their feasibility and scalability.

Various structural configurations have been explored for DPECs. Due to its compactness, the crossflow configuration was first adopted in the M-cycle cooler developed by Dr. Maisotsenko [13]. However, simulation analyses [22,42] have consistently shown that counterflow configurations outperform crossflow setups, resulting in significant improvements in both wet-bulb and dew-point effectiveness. Consequently, counterflow arrangements have emerged as the most effective layout, as demonstrated in studies by Zhan *et al.* [42], Chu *et al.* [43], Duan *et al.* [31], Riangvilaikul and Kumar [34], Wang *et al.* [38,39], and Lin *et al.* [27,36]. Counter-crossflow configurations, such as the one proposed by Jia *et al.* [33], aim to balance airflow stability and heat exchanger compactness, particularly for small-scale applications.

In terms of airflow direction, both horizontal and vertical arrangements have been investigated. Jradi and Riffat [26], Riangvilaikul and Kumar [34], Duan *et al.* [31], Xu *et al.* [32], and Liu *et al.* [40] developed DPECs featuring vertical channels and vertical air streams, while Khalid *et al.* [35], Chu *et al.* [43] and Wang *et al.* [29,38,39] employed horizontal channels and horizontal air streams. Zhu *et al.* [28] and Xu *et al.* [32] emphasized that water distribution across the wet channels plays a critical role in system performance; non-uniform wetting can lead to reduced effectiveness and operational instability. Xu *et al.* [32] further found that vertical airflow channels offer better gravity-assisted water distribution, while other studies noted improved compactness and integration potential with horizontal configurations.

Geometric parameters such as channel height, length, and spacing have been extensively investigated for performance optimization. Zhao *et al.* [21] studied flat-plate channel heat

exchangers and recommended channel heights below 6 mm and lengths approximately 200 times the height to maximize effectiveness while minimizing pressure drop. Liu *et al.* [19] explored alternative geometries—such as circular, triangular, and corrugated profiles—that may improve compactness and airflow uniformity. Baakeem *et al.* [24] specifically compared cylindrical and triangular profiles using both simulations and experiments, reporting notable performance improvements, but also highlighting increased complexity in water distribution management. Liu *et al.* [20] further examined perforated walls to enhance air mixing and heat transfer; however, these designs introduced greater structural complexity and a higher risk of bypass airflow.

Several studies have shown that material selection plays a crucial role in thermal conductivity and system durability [29,33]. Aluminum [38,39] or polymer-based materials [33,43] are commonly used in the fabrication of DPECs due to their favorable thermal and structural properties. To extend the applicability of DPECs in humid climates, multiple studies have proposed integrating desiccant dehumidification with dew-point cooling. Iyer *et al.* [25,41] investigated a desiccant-coated M-cycle cooler that enabled simultaneous cooling and moisture removal. Lao *et al.* [44] developed a combined sub-dew-point evaporative cooler capable of operating at low regeneration temperatures, making it potentially suitable for solar-assisted or waste-heat-driven systems. These hybrid approaches significantly enhance the latent cooling capacity of DPECs and mitigate performance limitations associated with high humidity.

Although previous studies have successfully enhanced DPEC performance through configuration and geometric improvements, these systems remain fundamentally limited by their single-stage configurations and relatively high working air ratios. Typically, 30–50% of the product air is diverted as working air, resulting in reduced delivered air volume and decreased energy efficiency [19,20,45]. A significant research gap remains in the development of compact, multi-stage systems capable of reducing the working air ratio without compromising cooling effectiveness. This need is particularly critical in regions with high cooling demands and constrained energy resources.

To date, no experimentally validated, compact, two-stage DPEC has been reported that achieves a high product air ratio (>88%) while maintaining superior wet-bulb and dew-point effectiveness under realistic operating conditions. This study addresses that gap by developing and experimentally evaluating a first-of-its-kind two-stage dew-point evaporative cooling system, as illustrated in Figure 1(b). The proposed system distributes the cooling load across two sequential stages: the first stage uses ambient air to pre-cool the process air close to its wet-bulb temperature; the second stage uses a small fraction of the cooled product air to further

reduce the process air temperature toward its dew-point. This staged approach significantly reduces the amount of diverted working air and enhances the system's overall energy efficiency.

The objectives of this study are to design, fabricate, and evaluate an innovative two-stage dew-point evaporative cooler that addresses the limitations of conventional single-stage systems through three key contributions: (1) development of a novel heat and mass exchanger with optimized airflow via carefully selected fan models; (2) systematic assessment of critical performance parameters (flow resistance, product air ratio) and their influence on cooling efficiency; and (3) comprehensive performance benchmarking against single-stage DPECs under subtropical summer conditions. By demonstrating the system's enhanced cooling effectiveness, energy efficiency, and compactness while establishing fan selection guidelines, this work provides a functional prototype and actionable insights for advancing sustainable cooling technologies.

## **2. Experimental**

### *2.1. Evaporative heat exchanger structure and fluid flow configuration*

A first-of-its-kind cooler was developed to implement the proposed two-stage cooling mechanism shown in Figure 1(c). A distinguishing feature of this cooler is that only a small fraction of the product air is redirected to serve as working air, enabling the majority of the product air to be cooled to temperatures below the wet-bulb temperature and approaching the dew-point temperature. This design significantly enhances energy efficiency and overall system performance. The prototype incorporates a heat and mass exchanger capable of managing four fluid flows—three air streams and one water flow—within a single compact unit. To ensure stable and high cooling performance, the materials and construction of the heat exchanger are optimized to maximize heat and mass transfer rates while minimizing flow resistance in the channels.

In this study, a novel heat and mass exchanger was meticulously designed and fabricated in a cross-flow configuration, which offers high compactness and minimal pressure drop. This arrangement is particularly well-suited for two-stage evaporative cooling, as it effectively facilitates the complex interaction of multiple fluid streams. The exchanger comprises a stacked assembly of 71 dry channels and 70 alternating wet channels. Flat aluminum sheets, with a thickness of 0.1 mm, were chosen as channel separators due to their superior thermal conductivity, low density, excellent durability, and waterproof properties. These characteristics ensure efficient heat transfer, ease of fabrication, and long-term stability, making aluminum an ideal material for evaporative cooling heat exchangers. The 0.1 mm thickness was specifically

selected to provide sufficient mechanical strength to prevent sagging while minimizing thermal resistance, as supported by reference [29,38,39].

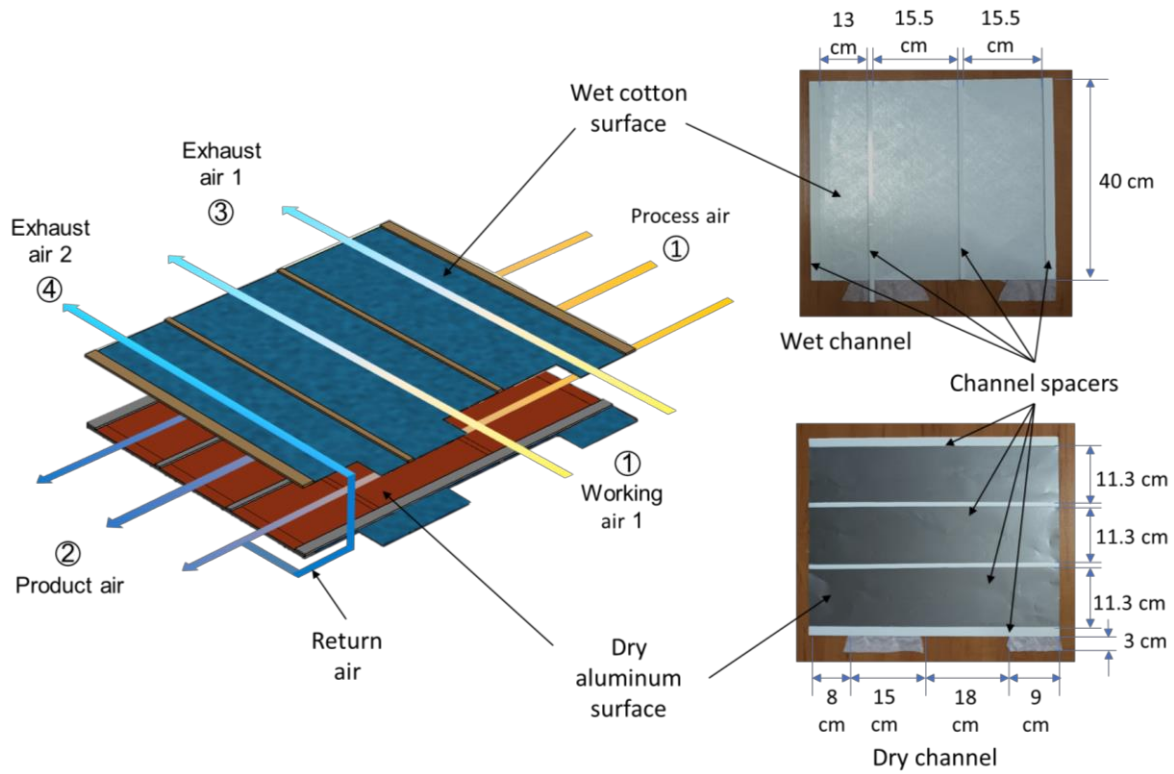


**Figure 2:** (a) channel's construction and fluid flow configuration of the two-stage indirect dew-point evaporative cooling exchanger; (b) pictorial views of the wet and dry channels with detailed dimensions.

To enhance water distribution within the wet channels, a thin layer of non-woven cotton fabric (0.22 mm thick) was securely bonded to the aluminum sheets using a waterproof adhesive. This fabric was chosen for its high capillary action, which ensures uniform water absorption and distribution along the wet channel walls, thereby maintaining consistent evaporative cooling performance. The total thickness of the channel separator in its dry state is 0.32 mm, which includes the aluminum sheet and the cotton fabric layer. Polypropylene spacers, each with a thickness of 3 mm, were employed to maintain precise separation between the channel layers, forming the air channels within the heat exchanger. Each channel measures 500 mm (length)  $\times$  400 mm (width)  $\times$  3 mm (height), inclusive of spacers. Detailed channel dimensions are provided in Figure 2(b).

The dry channel is formed between two dry aluminum surfaces and carries only the process airflow, as illustrated in Figure 2 and labeled as (1)  $\rightarrow$  (2). In contrast, the wet channel is formed between the cotton-coated surfaces of the channel separator and accommodates two separate working air flows: (1)  $\rightarrow$  (3) and (2)  $\rightarrow$  (4). A fraction of the product air ((2)) from the dry

channel is redirected into the wet channel to serve as working air, as depicted in Figure 2. The two working air flows ((1)→(3) and (2)→(4)) remain separated within the wet channel by non-permeable polypropylene spacers, preventing mixing. During operation, the process air flowing in the dry channel from (1) to (2) is first cooled by the working air from ambient air ((1)), followed by cooling with the product air ((2)) flowing in the adjacent wet channel. This sequential cooling process ensures that the product air is consistently cooled to temperatures below the wet-bulb temperature, thereby improving the overall cooling efficiency and performance of the system.

## 2.2. Fabrication of the cooler

The evaporative heat exchanger is assembled by alternating dry and wet channels in a structured arrangement, where each dry channel is positioned between two wet channels, and each wet channel is similarly sandwiched between two dry channels. The heat exchanger measures 40 cm in width, 50 cm in length, and 50 cm in height, as shown in Figure 3(a). To facilitate water distribution, two sections of the capillary cloth layer extend beyond the wet channels on the side where working air enters the heat exchanger. Water is supplied to these fabric extensions and spreads horizontally into the wet channels via capillary action. A water migration test was performed using a channel width of 40 cm to verify that the cotton cloth layer within the wet channels remains fully wetted and stable under airflow conditions. This configuration guarantees consistent and effective heat and mass transfer across the wet channels, improving the overall efficiency and reliability of the heat exchanger.

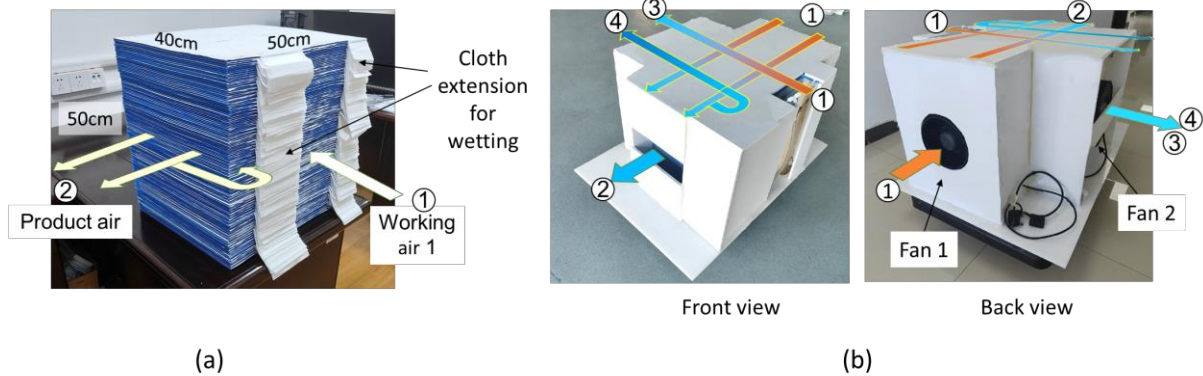


**Figure 3:** (a) heat exchanger and (b) cooler prototype with the airflow directions.

The heat exchanger is integrated with two identical fans and securely enclosed within a casing to form a complete cooling system, as illustrated in Figure 3(b). Fan 1, positioned at the back of the cooler, drives the process air ((2)) through the dry channels. Fan 2, located on the side of the cooler, draws working air ((1) and (2)) through the wet channels and expels the

exhausted air (③ and ④) to the ambient environment. This compact and efficient design enables optimal heat and mass transfer while minimizing the system's physical footprint, making it particularly suitable for applications where space is constrained. The strategic arrangement of the channels and casing ensures effective air management and maximizes cooling performance, contributing to the system's overall energy efficiency and operational effectiveness.

To assess the cooler's performance under different input air flow rates, two different fan models with distinct capacities were tested in combination with the heat exchanger. The technical parameters of these fans, detailing their performance characteristics, are presented in Table 1, while their static pressures versus flow rate characteristic curves are depicted in Figure 6.

**Table 1** Technical parameters of two types of fans in this work

| Specifications          | Fan model 1 <sup>(a)</sup> | Fan model 2 <sup>(b)</sup> |
|-------------------------|----------------------------|----------------------------|
| Types                   | Centrifugal Fan            | Axial Fan                  |
| Electricity             | Single phase, 220V, 50Hz   |                            |
| Power                   | 160 W                      | 75 W                       |
| Highest flowrate        | 1450 m <sup>3</sup> /h     | 829 m <sup>3</sup> /h      |
| Highest static pressure | 600 Pa                     | 120 Pa                     |
| Diameter                | 25 cm                      | 25 cm                      |
| Height                  | 6 cm                       | 8 cm                       |

<sup>(a)</sup> Suerke 250FLW2-A

<sup>(b)</sup> Weiguang YWF4E-250S-P92/25-G

Although both fan models share similar form factors, the centrifugal fan has a higher power rating, can deliver a greater volume flow rate, and generates higher static pressure compared to the axial fan. This comparative analysis allows for a thorough evaluation of the cooler's performance under various operating conditions, aiding in the identification of the most suitable fan to optimize efficiency and cooling capacity.

### 2.3. Cooler performance tests

The performance of the developed dew-point cooler was assessed under typical subtropical climatic conditions, ensuring the provision of 100% fresh, cooled air to the conditioned space. Air humidity and temperature for both the input and product airflows were monitored using a HOBO MX1101 data logger, which offers an accuracy of  $\pm 2\%$  for humidity and  $\pm 0.21^\circ\text{C}$  for temperature. Airspeed measurements were taken with a Testo 425 device, providing an

accuracy of  $\pm 0.03$  m/s. Static pressure was measured using a Testo 510, characterized by an accuracy of  $\pm 0.05$  hPa and a resolution of 1 Pa.

The experimental setup, designed to enable precise data collection and comprehensive performance analysis, is depicted in Figure 4. This arrangement ensured accurate monitoring of system parameters, providing reliable insights into the cooler's operational effectiveness under real-world conditions.

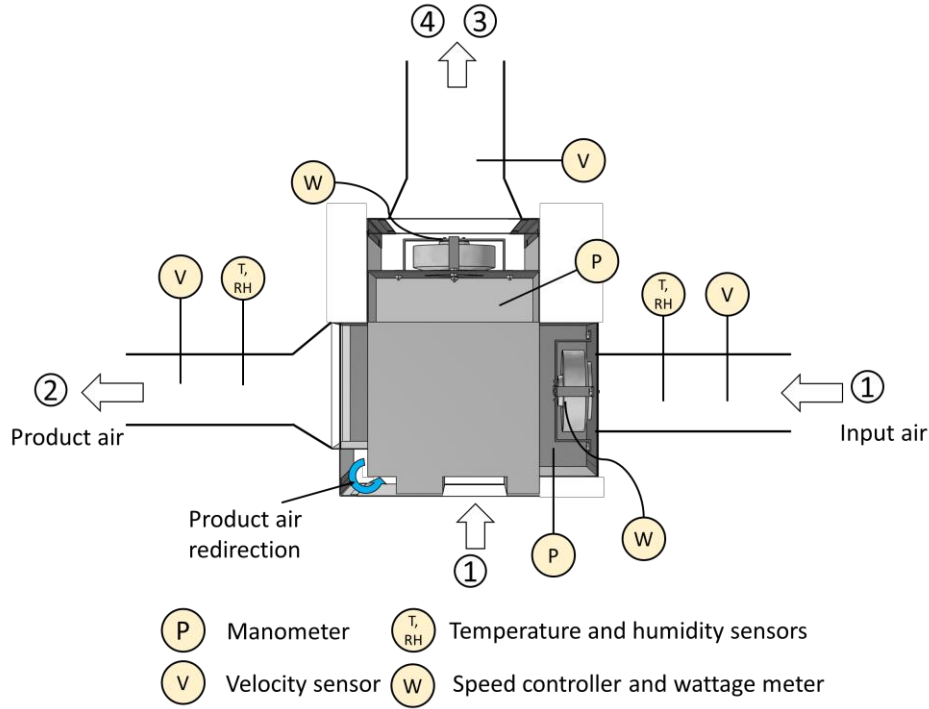


**Figure 4:** Experimental setup diagram

The speeds of the fans are regulated by changing the supply power voltage. The fan efficiency is derived from the volume flowrate,  $\dot{V}$ , pressure drop,  $P$ , and the electricity consumption of the fan,  $E$ , in the relation:

$$\eta = \frac{P\dot{V}}{E} \quad (1)$$

The product air ratio, denoted as  $r$ , is calculated based on the volume flow rates of the product air,  $\dot{V}_{pro}$  and the input air,  $\dot{V}_{in}$ :

$$r = \frac{\dot{V}_{pro}}{\dot{V}_{in}} \quad (2)$$

Two key metrics are used to assess the cooling performance of the fabricated dew-point cooler, namely the wet-bulb effectiveness ( $\xi_{wb}$ ) and dew-point effectiveness ( $\xi_{dp}$ ):

$$\xi_{wb} = \frac{T_{in} - T_{pro}}{T_{in} - T_{wb}} \times 100\% \quad (3a)$$

$$\xi_{dp} = \frac{T_{in} - T_{pro}}{T_{in} - T_{dp}} \times 100\% \quad (3b)$$

The cooler's cooling capacity,  $Q$ , is derived by:

$$Q = (T_{in} - T_{pro}) \times \dot{m}_{pro} \times 1.006 \frac{kJ}{kg \cdot K} \quad (4)$$

The cooler's energy efficiency is quantified using the cooling coefficient of performance (COP), which is the ratio of the cooling capacity,  $Q$ , to the fan electricity consumption,  $E$ . This metric is expressed as:

$$COP = \frac{Q}{E} \quad (5)$$

The mean heat flux through the channel plates is derived as:

$$q = \frac{(T_{in} - T_{pro}) \times \dot{m}_{in} \times 1.006 \frac{kJ}{kg \cdot K}}{A} \quad (6)$$

#### 2.4. Propagation of error

An analysis of the uncertainty propagation was conducted to assess the reliability of the measurement, following Equation (7):

$$e = \pm \sqrt{\sum_{i=1}^n \left( \frac{\partial f}{\partial x_i} \right)^2 (\Delta x_i)^2} \quad (7)$$

### 3. Results and discussion

#### 3.1. Flow resistance characteristics

The process air and working air flowing through the heat exchanger's channels encounter flow resistance due to frictional forces from the channel walls. Lower pressure drops across the channels reduce the electricity consumption of the fans. The pressure drops in both wet and dry channels are measured at varying fan speeds. As illustrated in Figure 5(a), the pressure drops in both channels increase significantly with higher airflow rates. The pressure drop in the wet channel is roughly double that observed in the dry channel. This is due to the narrower entrance area of the wet channel, which is caused by the cloth layer extensions for water flow, as illustrated in Figures 2 and 3(a). Consequently, the volume flow rate of the working air is lower than that of the process air when two fans of the same model are used. At maximum operating voltage (220 V), the centrifugal fan (Model 1) delivers air flow rates of 1221 m<sup>3</sup>/h (yielding an average air velocity of 4.68 m/s) and 974 m<sup>3</sup>/h (2.08 m/s average velocity) in the dry and wet channels, respectively. By comparison, the axial fan (Model 2) provides flow rates of 542 m<sup>3</sup>/h (2.93 m/s average velocity) and 400 m<sup>3</sup>/h (1.20 m/s average velocity) for the same channels, respectively. Table 2 summarizes the ranges of air flow rates and velocities in the dry and wet channels for the two fan models. The efficiencies of the two fan models are determined by Equation (1) and shown in Figure 5(b). It is observed that fan efficiency is proportional to

fan speed. The highest efficiency of fan Model 1 is 24% at its maximal flow rate of 1239 m<sup>3</sup>/h, while that of fan Model 2 is 15% at 542 m<sup>3</sup>/h..

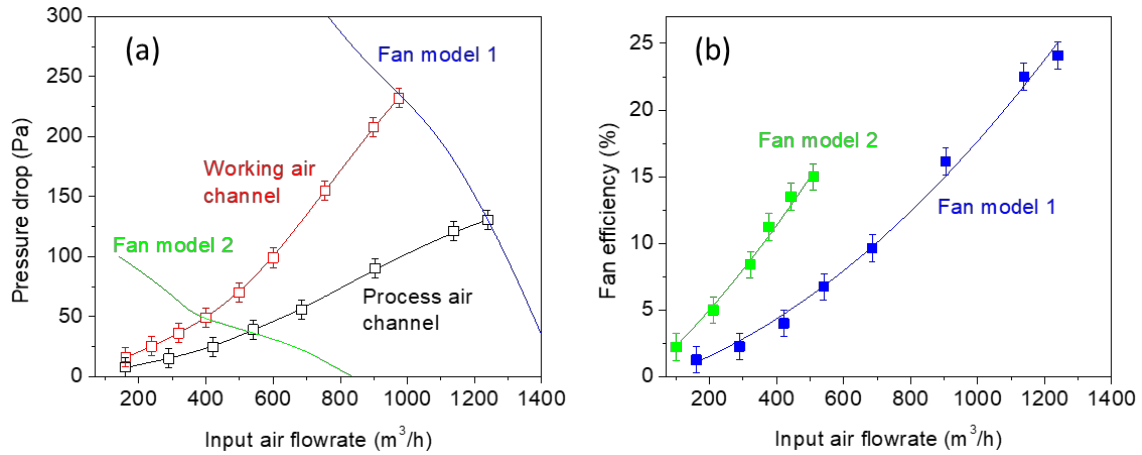


**Figure 5:** (a) pressure drops in the heat exchanger channels and fan curves, and (b) efficiencies of the two fan models

**Table 2.** Air flow rate ranges and corresponding average velocities in dry and wet channels for two fan models

|                                                        | Fan model 1                                     | Fan model 2                                    |
|--------------------------------------------------------|-------------------------------------------------|------------------------------------------------|
| Applied potentials                                     | 80-220 V                                        |                                                |
| Air flowrate and average air speed in dry channels     | 159-1229 m <sup>3</sup> /h<br>and 0.61-4.75 m/s | 100-542 m <sup>3</sup> /h<br>and 0.38-2.08 m/s |
| Air flowrate and average air speed in wet dry channels | 100-974 m <sup>3</sup> /h<br>and 0.30-2.93 m/s  | 75-400 m <sup>3</sup> /h<br>and 0.22-1.20 m/s  |

### 3.2. Product air ratio

The product air ratio of the developed cooler, a critical parameter influencing its performance, is defined as the ratio of the product air volume flow rate to the total input air volume flow rate, as expressed in Equation (2). This ratio was determined by systematically varying the input process air flow rates, achieved by adjusting the power voltage applied to Fan 1 (ranging from 80 V to 220 V), while Fan 2 operates at its maximum speed with a 220 V input. As illustrated in Figure 6, the output product air flow rate rises from 145 m<sup>3</sup>/h to 1091 m<sup>3</sup>/h as the input process air flow rate is adjusted from 159 m<sup>3</sup>/h to 1221 m<sup>3</sup>/h, demonstrating a nearly linear relationship. Despite these variations, the product air ratio remains stable, ranging from 88% to 90.5%, indicating consistent performance across the tested conditions. This stability underscores the cooler's ability to maintain efficiency across a wide range of operating

conditions, ensuring reliable functionality.

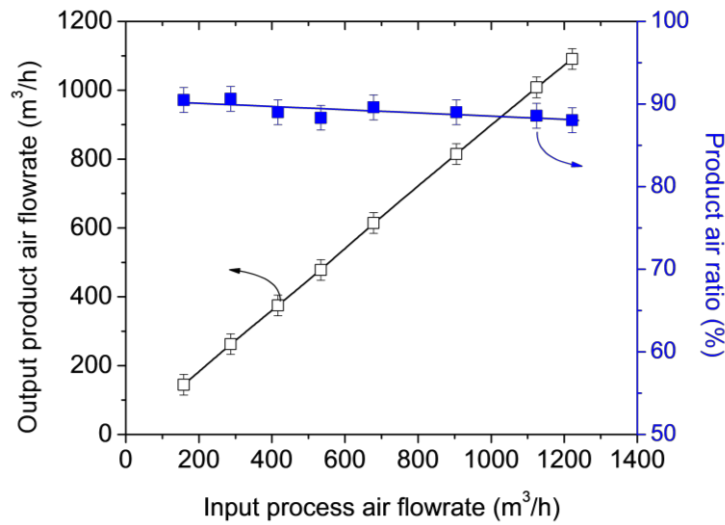


**Figure 6:** Product air flowrate and product air ratio as the input process air flowrate is varied.

### 3.3. Cooler performance with centrifugal fans used

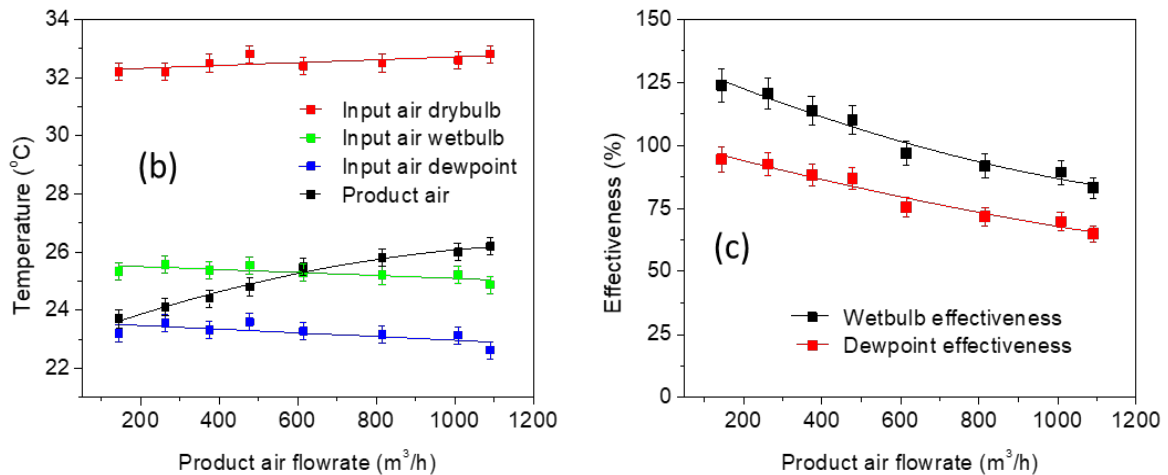


**Figure 7:** Changes in (a) the air temperature and (b) the wet-bulb and dew-bulb effectiveness as the product airflow rate is modified.

The efficacy of the cooling system is assessed using two identical fans (Model 1) to drive the process air and working air through the developed heat exchanger. Testing is conducted under outdoor air conditions of  $32.5 \pm 0.5^{\circ}\text{C}$  dry-bulb temperature and  $25 \pm 0.5^{\circ}\text{C}$  wet-bulb temperature (corresponding to a relative humidity of  $55 \pm 2\%$ ) on a sunny summer day in the subtropical climate of Zhenjiang City, China. During the test, the speed of the process air fan is varied by adjusting its input voltage from 80 V to 220 V, while the speed of the working air fan is set to the maximum at 220 V. The temperatures of the input process air and output product air are recorded and graphically represented as dependent variables relative to the

volume flow rate of the product air, as illustrated in Figure 7(a). The results show that the product air temperature decreases with lower product air flow rates, which results from the longer retention time of the process air in the dry channels at lower flow rates. The wet-bulb and dew-point effectiveness of the cooler are evaluated and shown in Figure 7(b). The wet-bulb effectiveness exhibits a significant reduction from 123% to 88%, while the dew-point effectiveness demonstrates a notable decline from 95% to 68%, as the volumetric flow rate of the product air escalates from 145 m<sup>3</sup>/h to 1091 m<sup>3</sup>/h. A similar trade-off between airflow rate and cooling efficiency is also reported in other dew-point coolers [26,32,33,35,38].

It is particularly significant to highlight that throughout the testing process, the product air ratio remains consistently at 88–90%, as shown in Figure 6. Although only 10–12% of the product air is redirected to the wet channel to function as working air, the system remains capable of cooling the process air to temperatures below the wet-bulb level and nearing the dew-point temperature, particularly when the product air flow rate remains under 600 m<sup>3</sup>/h. Under similar conditions, conventional dew-point coolers require 40–50% of the product air as working air, leading to a scenario where only 50–60% of the total product air is available for practical use [21,28,32,34,38,40]. In comparison, this cooler improves the product air flow rate by 30–40%, highlighting the superior cooling performance of the proposed two-stage cooling mechanism depicted in Figure 1.

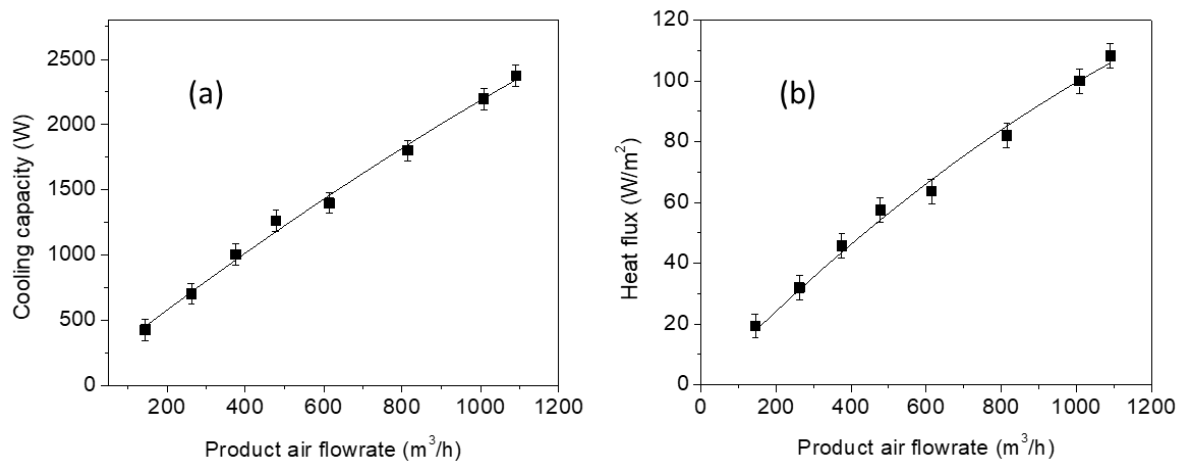


**Figure 8:** Changes in (a) the cooling capacity and (b) the heat flux as the product air flowrate is modified.

The cooling capacity and average heat flux are calculated using Equations (4) and (6), respectively, and are shown in Figure 8. Although the temperature reduction ( $T_{in} - T_{pro}$ ) is smaller, the cooling capacity increases at higher product air flow rates. This phenomenon occurs because the rise in the product air flow rate has a more pronounced impact than the

corresponding decrease in temperature reduction. As illustrated in Figure 8(a), increasing the product air flow rate from 145 m<sup>3</sup>/h to 1091 m<sup>3</sup>/h results in a rise in cooling capacity from 425 W to 2375 W. Likewise, Figure 8(b) demonstrates that as the product air volume flow rate grows within the tested range, the mean heat flux increases from 19.3 W/m<sup>2</sup>·K to 108 W/m<sup>2</sup>·K. This demonstrates the significant impact of product air flow rate on heat transfer and the overall energy efficiency of the system.

The energy consumption of the working-air and process-air fans is measured and recorded. These data points are then graphically represented as functions of the product air flow rate, as illustrated in Figure 9(a). The working air fan, set to its highest speed, maintains a constant power consumption of 232 W. In contrast, the power consumption of the process air fan, which is regulated by adjusting the applied voltage, increases from 26 W to 186 W across the range of product air flow rates. At the highest flow rate, the power consumption of the working air fan exceeds that of the process air fan. This is consistent with the higher flow resistance observed in the wet channel, as illustrated in Figure 5.

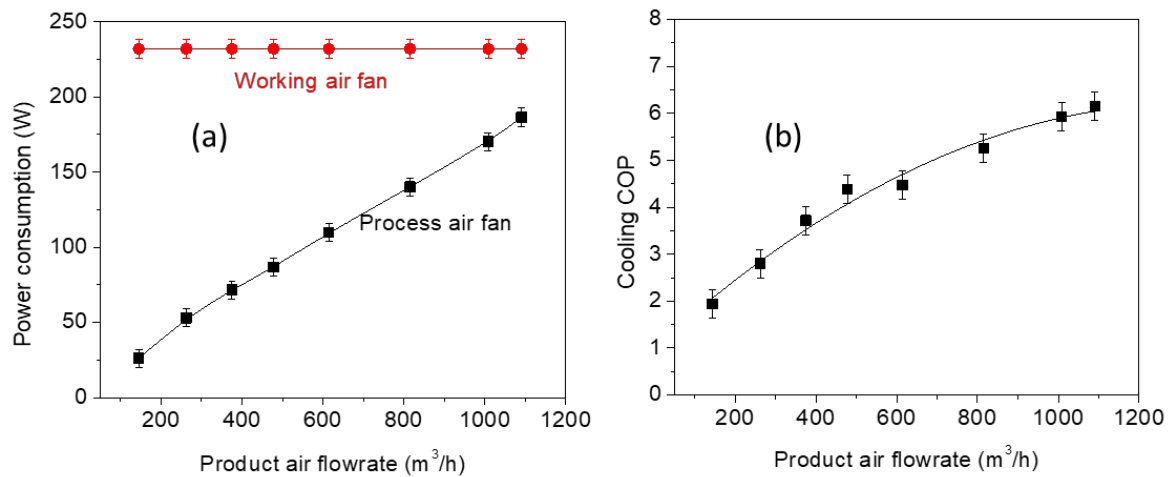


**Figure 9:** Changes in (a) the power consumption and (b) the cooling COP as the product air flowrate is modified.

The cooling coefficient of performance (COP) of the cooler is the ratio of cooling capacity to the total power consumption of the fans, as expressed in Equation (5) and depicted in Figure 9(b). Although the fans consume more electricity to achieve higher product air flow rates, as illustrated in Figure 9(a), the cooling COP improves across the range of product air flow rates. This indicates that the enhancement in cooling capacity more than compensates for the corresponding increase in power consumption as the airflow rate escalates. It is noteworthy

that the COP ranges from 1.9 to 6.1 within the flow rate range, despite the fan efficiency being lower than 24%, as shown in Figure 5(b).

### 3.4. Cooler performance with axial fans used

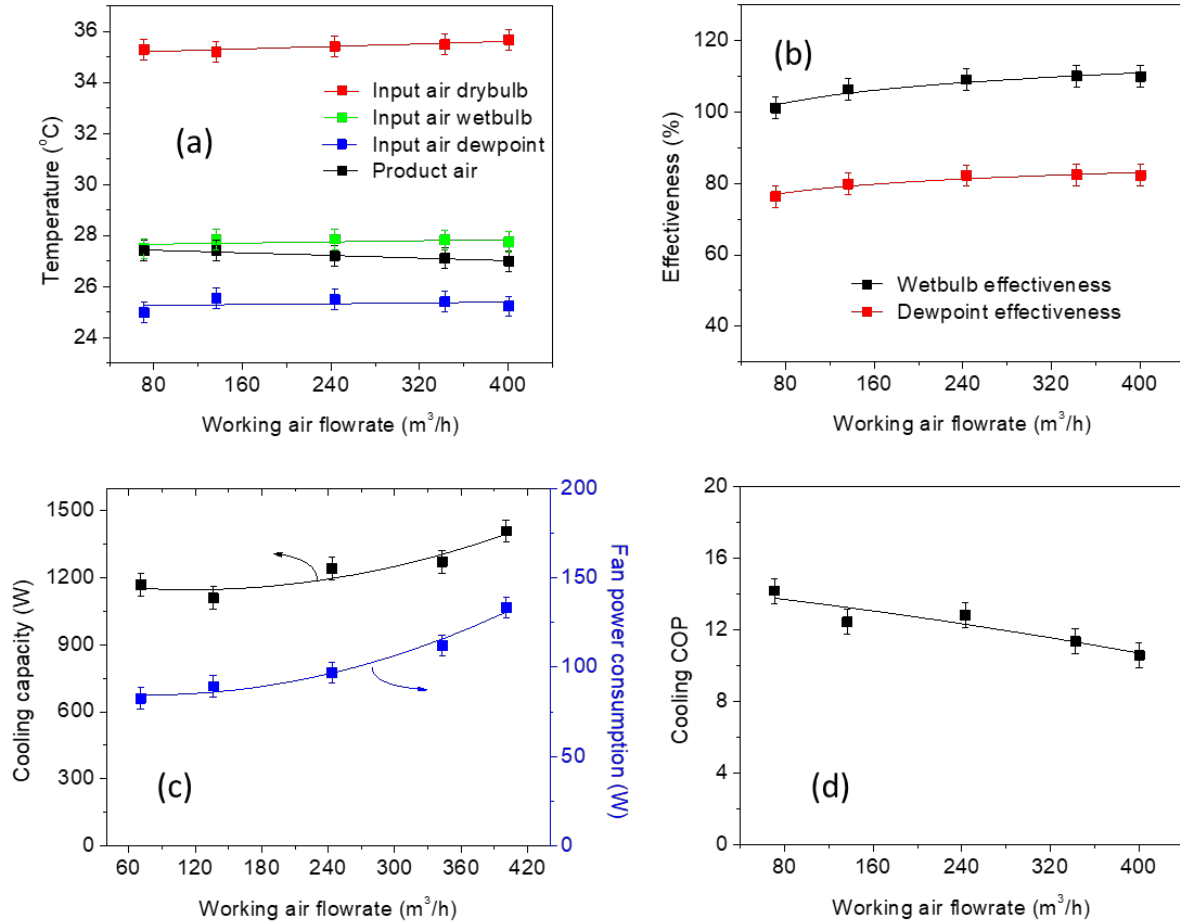


**Figure 10:** Changes in (a) the air temperature, (b) the wet-bulb and dew-bulb effectiveness, (c) the cooling capacity and fan power consumption, (d) the cooling COP as the working air flowrate is modified.

In this test, the two fans of the cooler are replaced with two identical axial fans (Model 2). The process air fan is set to its maximum speed with an applied voltage of 220 V, while the working air fan speed is adjusted by varying the applied voltage from 100 V to 220 V. The cooler is evaluated under tropical environmental conditions characterized by high temperatures and elevated humidity levels, typical of a sunny summer day, with an ambient air dry-bulb temperature of  $35.5 \pm 0.3^\circ\text{C}$  and a wet-bulb temperature of  $27.6 \pm 0.3^\circ\text{C}$  (corresponding to a relative humidity of  $55 \pm 2\%$ ). The temperatures of the input air and product air are graphically represented as functions of the working air flow rate, as depicted in Figure 10(a). The data show that the input air is consistently cooled to temperatures below the wet-bulb temperature

1 across the entire range of airflow rates. Higher working air flow rates improve the cooling  
2 performance, resulting in lower product air temperatures. As illustrated in Figure 10(b), the  
3 wet-bulb effectiveness shows an upward trend, increasing from 101–110%, while the dew-  
4 point effectiveness also rises, climbing from 76% to 82% with the increase in working air flow  
5 rate.

6 Figure 10(c) depicts the cooling capacity and fan power consumption as functions of the  
7 working air flow rate. As the working air flow rate increases, the cooling capacity rises due to  
8 a higher temperature reduction. Similarly, power consumption increases with the higher  
9 working air flow rate. However, the rate of increase in power consumption exceeds that of the  
10 cooling capacity, leading to a decrease in the cooling COP across the range of working air flow  
11 rates, as shown in Figure 10(d). It is important to point out that a high cooling COP ranging  
12 from 14.2 to 10.6 is attained, even though the fan efficiency is lower than 15%, as shown in  
13 Figure 5(b). This suggests that there is still significant potential for COP improvement when  
14 the developed heat and mass exchanger is coupled with a more efficient fan.

15 Notably, at their highest operating speeds, axial fans produce lower airflow rates and air  
16 speeds compared to centrifugal fans yet achieve lower product air temperatures. Although they  
17 deliver a lower cooling capacity, they offer higher energy efficiency. This underscores the  
18 importance of selecting the appropriate fan type to pair with the dew-point evaporative cooling  
19 heat exchanger. Axial fans are the better choice for applications requiring lower product air  
20 temperatures and higher energy efficiency, while centrifugal fans are more suitable for  
21 applications that demand higher airflow rates and faster air speeds.

### 22 23 *3.5. Comparison with recently reported dew-point coolers*

24 Table 3 provides a comparative analysis of the dew-point cooler developed in this work  
25 alongside recently designed dew-point coolers, all evaluated under comparable testing  
26 conditions. Given that the energy efficiency (COP) and cooling effectiveness (wet-bulb  
27 effectiveness and dew-point effectiveness) of a dew-point cooler are significantly affected by  
28 the dynamics of air flow interaction with the channel separators of the heat exchanger, it is  
29 crucial to compare these parameters in the context of airflow rates and heat exchanger  
30 dimensions. The cooler compactness is defined as the ratio of the product air volume flow rate  
31 to the heat exchanger volume.

**Table 3:** A comparative analysis of the developed cooler developed with several single-stage dew-point coolers reported recently.

| Reference / cooling mechanism | Input air conditions                      | Product air flowrate (m <sup>3</sup> /h) | Heat exchanger volume (m <sup>3</sup> ) | Compactness (m <sup>3</sup> /h/m <sup>3</sup> ) | Product air ratio | COP       | $\xi_{dp}$ (%) | $\xi_{wb}$ (%) |
|-------------------------------|-------------------------------------------|------------------------------------------|-----------------------------------------|-------------------------------------------------|-------------------|-----------|----------------|----------------|
| This work / Two-stage         | $T_{in}$ : 32-35°C,<br>$T_{wb}$ : 25-27°C | 145-1090                                 | 0.1                                     | 1450-10900                                      | 0.88-0.9          | 1.9-6.1   | 68-95          | 88-123         |
|                               |                                           | 500                                      |                                         | 5000                                            |                   | 10.6-14.2 | 76-82          | 101-110        |
| [38] / Single-stage           | $T_{in}$ : 32-33°C,<br>$T_{wb}$ : 25-26°C | 175-480                                  | 0.09                                    | 1944-5333                                       | 0.5               | 6-51      | 63-84          | 85-113         |
| [40] / Single-stage           | $T_{in}$ : 25.4-38.7°C, RH: 15.9-34.6%    | 974-1323                                 | 0.756                                   | 1288-1750                                       | 0.46-0.64         | 20-36.6   | 42.9-77.3      | 74.9-107.5     |
| [32] / Single-stage           | $T_{in}$ : 26.0°C,<br>$T_{wb}$ : 19.6°C   | 750                                      | 0.286                                   | 2622                                            | 0.64              | 18.1      | 69             | 104            |
|                               | $T_{in}$ : 30.5°C,<br>$T_{wb}$ : 26.2°C   |                                          |                                         |                                                 |                   | 16.5      | 95             | 128            |
| [34] / Single-stage           | $T_{in}$ : 25-45°C,<br>HR: 7-26g/kg       | 1.45-5.79                                | 0.00432                                 | 335-1340                                        | 0.67              | -         | 58-84          | 92-114         |
| [33] / Single-stage           | $T_{in}$ : 29.5-35.5°C, HR: 13-19g/kg     | 14-34                                    | 0.033                                   | 424-1030                                        | 0.6               | 10.1-13.8 | -              | 58.1-71.1      |

As indicated in Table 3, the cooling performance of the developed cooler in terms of wet-bulb effectiveness and dew-point effectiveness is on par with those of other reported coolers. Nonetheless, while maintaining comparable cooling performance, the cooler in this study exhibits a superior coefficient of performance and compactness. These advancements are primarily due to its innovative design and refined airflow configuration, which minimize pressure drop and improve airflow uniformity. As a result, the cooler offers superior overall performance and reduced space requirements, making it a more efficient and compact solution compared to other alternatives.

### 3.6. Cooler performance in actual operation conditions

The developed dew-point cooler, equipped with two axial fans, is used to supply cooled and fresh air to an enclosed space through a building window, as shown in Figures 11(a) and 11(b). The cooling performance is evaluated under real-world operating conditions, specifically on sunny days, to accurately reflect the system's effectiveness in practical scenarios. The sensible cooling process, which transitions the input air to the output product air, is graphically represented on a psychrometric chart in Figure 11(c), with the cooling COP indicated by the color scale of the data points. The test results show that, despite input air temperature

fluctuations between 29°C and 36°C, the output product air temperature is consistently maintained within a tightly controlled range of 23.2°C to 26.5°C. Notably, the cooling COP is higher when the air is hotter and less humid. On hot and dry days, the cooling COP reaches 15–17, which is significantly higher than the typical COP of commercial air conditioners. Moreover, if more efficient fans are used, the cooler can achieve even higher cooling COP, indicating its potential as a sustainable solution to air conditioning challenges.

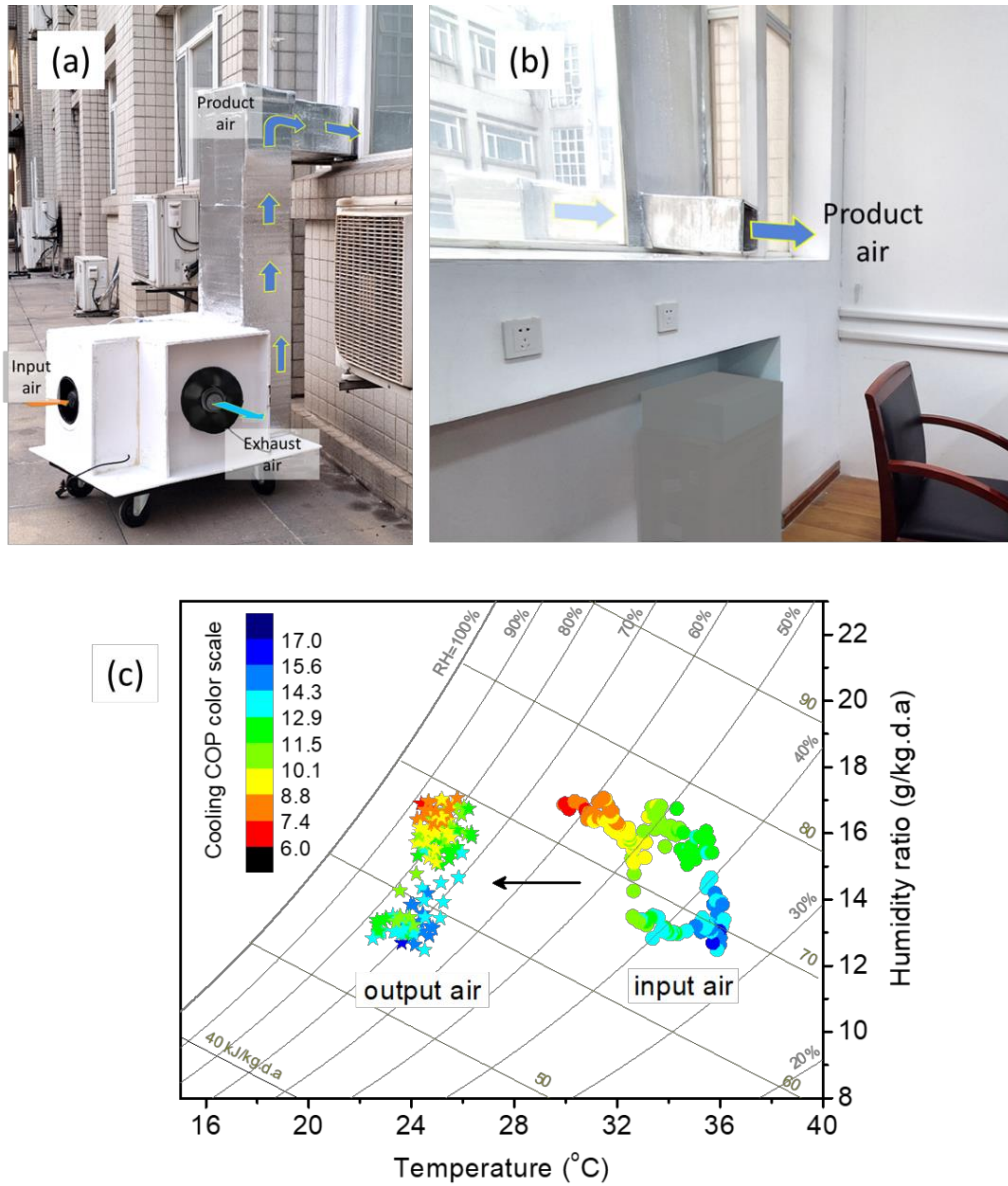


**Figure 11:** (a) outdoor deployment of the developed cooler in actual operation, (b) indoor product air supply, and (c) sensible cooling from the input ambient temperatures to the output product air temperatures.

#### 4. Conclusion

This study developed and experimentally validated a novel two-stage dew-point evaporative cooler (DPEC) aimed at overcoming the limitations of conventional single-stage systems. The main findings are summarized below:

- 1) A first-of-its-kind two-stage DPEC was successfully designed and tested. The system distributes the cooling load across two sequential stages, significantly reducing the working air ratio while improving energy efficiency.
- 2) A compact heat and mass exchanger accommodating four fluid flows (three air streams and one water flow) was developed, enabling efficient heat and mass transfer in a small footprint.
- 3) The prototype achieved a product air ratio of 88–90%, substantially higher than typical single-stage DPECs, which commonly divert 30–50% of the product air as working air.
- 4) Cooling performance was significantly enhanced:
  - a. Maximum wet-bulb effectiveness reached 129%,
  - b. Dew-point effectiveness reached 99%,
  - c. Coefficient of performance (COP) ranged from 10.6 to 14.2 under subtropical conditions, greatly surpassing conventional evaporative and vapor-compression systems.
- 5) Quantitative analysis showed that the proposed two-stage configuration effectively addresses the long-standing challenges of high working air ratios, limited cooling capacity, and low energy efficiency in single-stage DPECs.
- 6) Sensitivity analyses on fan models and flow resistances confirmed the influence of operational conditions on performance, providing practical insights for further system optimization.

The proposed two-stage DPEC represents a significant advancement in energy-efficient cooling technologies, especially for regions with high cooling demands and limited energy resources. Its high performance, compactness, and low energy consumption make it a viable alternative to traditional air conditioning systems.

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## **A first-of-its-kind two-stage dew-point evaporative cooler with high energy efficiency and compact design**

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