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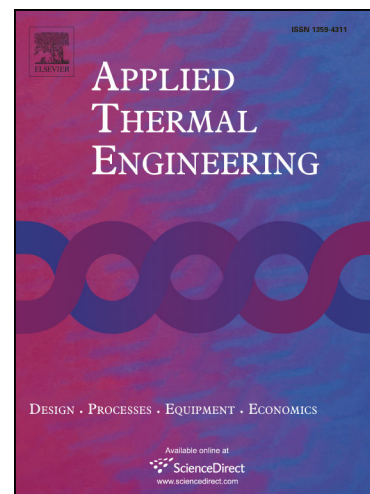
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Energy-Efficient Cooling Beyond M-Cycle: Development and Evaluation of a Two-Stage Dew-Point Evaporative Cooler

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Abstract

The Maisotsenko cycle (M-cycle) enables indirect evaporative cooling of air to near the dew point, offering a low-energy alternative to vapor-compression cooling. However, conventional M-cycle coolers operate as single-stage systems and are limited by high working air ratios (30–50%), low product air output, and constrained energy efficiency. To address these limitations, a novel two-stage dew-point evaporative cooler (DPEC) was recently proposed, employing sequential wet-bulb and dew-point cooling stages to redistribute the thermal load and enhance performance. This study presents a comprehensive numerical analysis of the two-stage DPEC using a validated three-dimensional COMSOL-based model. The model simulates full-scale air channels under realistic hot and humid conditions and enables direct comparisons with conventional single-stage wet-bulb and M-cycle systems. Results show that the two-stage DPEC achieves a high product air ratio (~90%) and dew-point effectiveness up to 95%, significantly outperforming traditional designs. Key performance metrics—such as cooling capacity, COP, and airflow utilization—are examined across varying climates. The findings confirm the thermodynamic advantages of load partitioning and highlight the system's potential as a scalable, energy-efficient solution for sustainable cooling in hot regions.

Keywords: Dew-point evaporative cooling, Two-stage cooling system, Heat and mass exchanger, Energy-efficient air conditioning, Product air ratio, Low-energy HVAC.

Nomenclature

c	Concentration, mol/m ³
C	Specific heat capacity at constant pressure, kJ/(kg · K)
D	Diffusivity, m ² /s
E	Fan power, W
$\mathbf{F}$	Volume force vector, N/m ³
$\mathbf{g}$	Mass diffusion flux, kg/m ²
G	Moisture source, kg/(m ³ ·s)
$\mathbf{I}$	Identity tensor
h	Specific enthalpy of incoming moist air, kJ/kg
H	Cooling capacity, W
k	Thermal conductivity, W/(m · K)
$\mathbf{K}$	Viscous stress tensor
L	Latent heat of vaporization, kJ/kg
$\dot{m}$	Mass flowrate, kg/s
M	Molecular mass, kg/mol
$\mathbf{n}$	Unit normal vector to the surface
P	Pressure, Pa
Q	Heat sources, W/m ³

$\mathbf{q}$	Heat flux vector, W/m ²
r	Product air ratio
R	Evaporation rate factor, m/s
T	Air temperature, K
$\mathbf{u}$	Air velocity vector, m/s
U	Velocity value, m/s
$\dot{V}$	Air volume flowrate, m ³ /h

Greek symbols

η	Efficiency of used fan
μ	Dynamic viscosity, Pa · s
ρ	Air density (kg/m ³)
ξ	Cooling effectiveness, %
ϕ	Relative humidity, %

Subscripts

amb	Ambient
dp	Dew-point
$evap$	Evaporation
in	Input

out Output

pro Product

wb Wet-bulb

sat Saturation state

v Water vapor

Abbreviations

COP Coefficient of performance

DPEC Dew-point evaporative cooler

IEC Indirect evaporative cooler

HR Humidity ratio

RH Relative humidity

1. Introduction

Air conditioning has become an indispensable element of modern infrastructure, providing essential thermal comfort and environmental control across diverse sectors [1]. It significantly boosts human productivity and well-being, particularly in regions with extreme climates [2–4], and is critical for safeguarding temperature-sensitive assets such as pharmaceuticals, food, and electronic storage [5]. However, the pervasive reliance on conventional air conditioning systems, predominantly those based on vapor-compression cycles, carries a substantial

environmental burden. These systems often utilize refrigerants that are potent greenhouse gases, contributing directly to global warming and ozone depletion. Furthermore, their considerable electricity consumption, largely sourced from fossil fuels, exacerbates their ecological footprint [6].

Globally, the demand for air conditioning is immense, accounting for nearly half of the energy consumed in the building sector in developed nations, a figure that escalates in tropical and subtropical zones [7]. This escalating demand strains energy grids, intensifies dependence on finite non-renewable resources, and creates significant peak load challenges during hot seasons, complicating efforts towards sustainable energy management. Beyond energy concerns, inadequately designed air conditioning systems can compromise indoor air quality [8], leading to ventilation deficiencies, pollutant accumulation, and respiratory issues, a concern profoundly underscored by the recent coronavirus disease 2019 pandemic [9]. The intricate balance between achieving thermal comfort, optimizing energy efficiency, and ensuring adequate ventilation remains a paramount challenge in the design and operation of modern cooling systems.

In response to these challenges, evaporative coolers have emerged as a compelling energy-efficient and environmentally conscious alternative to traditional air conditioning. Their fundamental principle involves leveraging the latent heat of water evaporation to achieve cooling [10]. Compared to conventional air conditioners, evaporative coolers offer distinct advantages: superior energy efficiency, reduced initial and operational costs, and simpler installation and maintenance. Moreover, their environmental sustainability is notable, as they operate without releasing harmful chemicals or excess heat into the atmosphere [10,11].

Evaporative coolers are broadly classified into two main types: direct and indirect. Direct evaporative coolers cool air through direct contact with water, which, while lowering temperature, concurrently increases air humidity. In contrast, indirect evaporative coolers (IECs) ingeniously separate the air streams into wet and dry channels, facilitating heat exchange without introducing moisture into the conditioned air, thereby preserving its original humidity levels. This characteristic makes IECs suitable for applications where maintaining low humidity is crucial. The performance of conventional wet-bulb IECs is inherently limited by the wet-bulb temperature of the incoming air, as thermodynamic principles dictate that the cooled air cannot fall below this threshold [12–15].

A significant leap in evaporative cooling technology was pioneered by Dr. Maisotsenko, leading to the development of advanced dew-point evaporative coolers (DPECs), also known as M-cycle cooling systems. This innovative technology integrates heat transfer with water evaporation to achieve enhanced cooling performance, enabling air to be cooled below its wet-bulb temperature, approaching its dew-point temperature [10,11]. This is achieved by redirecting a portion of the pre-cooled air from the dry channel to serve as "working air" in the wet channel [12–15].

Research on single-stage dew-point evaporative coolers (DPECs) has evolved significantly over the past two decades, encompassing simulation and experimental, approaches. Simulation studies range from simplified one-dimensional [16,17] and two-dimensional [18–24] models to advanced three-dimensional [25–27] computational fluid dynamics simulations, enabling detailed analysis of design and operating parameters such as inlet air conditions, working ratios, channel dimensions, and material properties. Experimental investigations validate these models and assess practical performance. Notable prototypes include the commercial Coolerado unit [10] and academic developments by Riangvilaikul and Kumar [28], Xu *et al.*

[29], Chu *et al.* [30], and others [31–35], focusing on effectiveness, cooling capacity, and COP under diverse climates.

To improve performance, researchers have explored enhancements in channel configuration, materials, and airflow arrangements. Early DPECs typically employed crossflow channels due to their simplicity and low pressure drop [10]. However, counterflow designs are now preferred for their superior heat and mass transfer efficiency. For example, Zhan *et al.* [19,36] reported 15–23% higher effectiveness and 20% greater cooling capacity with counterflow over crossflow. Recent innovations, such as those by Wang *et al.* [34,35], have further refined airflow paths by eliminating abrupt direction changes, reducing pressure drop and improving compactness. Beyond conventional geometries, studies have investigated novel configurations such as counter-crossflow (Jia *et al.*, [37]) and shell-and-tube designs (Sulaiman *et al.* [38]), offering improvements in compactness and cooling output.

Material selection also plays a crucial role. While early systems used flat aluminum plates, newer designs utilize corrugated surfaces and composite materials to enhance heat transfer and reduce resistance. Corrugated exchangers, as shown by Xu *et al.* [29], achieved up to 37% higher effectiveness and over 50% lower pressure drop. Jia *et al.* [37] and Chu *et al.* [30] demonstrated improved performance using polystyrene, nylon fiber, and plant fiber composites.

More recent studies emphasize functional coatings and integrated systems. Iyer *et al.* [39] introduced aluminum plates coated with silica gel and cotton fabric to enable simultaneous evaporative cooling and desiccant dehumidification. Marco Lao *et al.* [40] developed a combined sorption-dehumidification and evaporative cooling system, achieving deep dehumidification and sub-dew-point cooling with effectiveness values exceeding 2.8, even at low regeneration temperatures ($\sim 40^\circ\text{C}$). Such systems represent a significant step toward integrated, low-energy, and climate-resilient air conditioning technologies.

Despite the promising potential of DPEC as an energy-efficient and environmentally sustainable alternative to conventional air conditioning, most existing systems are limited to single-stage configurations. These systems typically rely on high working air ratios (30–50%), which significantly reduce the volume of usable product air and result in diminished cooling capacity and energy efficiency. Moreover, while previous studies have explored variations in channel geometry, materials, and airflow configurations, few have addressed the fundamental structural limitation imposed by the single-stage design. A critical research gap persists in the absence of multi-stage DPEC systems that effectively mitigate this working air requirement while simultaneously achieving higher dew-point effectiveness and cooling capacity. To date, no experimentally validated system that successfully overcomes the intrinsic energy inefficiencies and airflow trade-offs characteristic of single-stage systems.

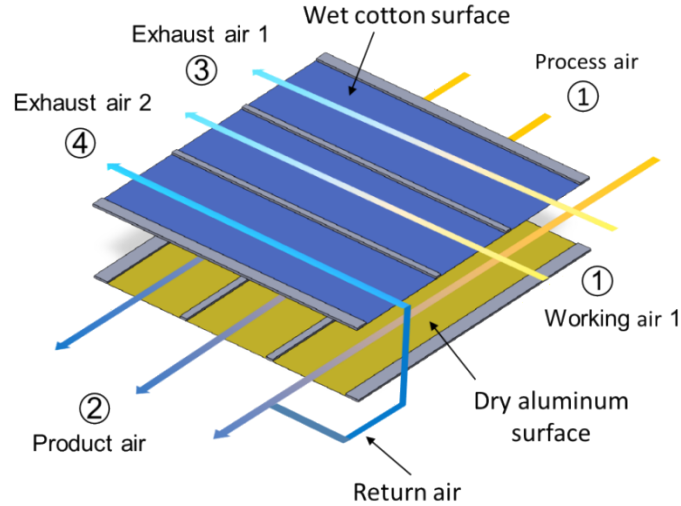


Figure 1: Air flow configuration of the developed two-stage dew-point evaporative cooler [41].

To overcome these limitations, Z. Chen *et al.* [41] recently proposed and validated a novel DPEC system based on the configuration shown in Figure 1. A system qualifies as two-stage when it meets three core criteria: (1) two distinct and sequential cooling processes, (2) thermodynamically independent stages with separate wet surfaces and airflow paths, and (3) partial thermal load distribution across stages. The first stage in this system is a conventional IEC, which pre-cools the air using ambient working air. The second stage is a conventional M-cycle, which further cools the air toward the dew point. Each stage operates independently with its own wetted channels and working air stream, while internally partitioned flow channels enable modular operation. This compact configuration not only meets the criteria for a two-stage DPEC system but also provides a scalable foundation for future multi-stage development.

Markedly, the first stage plays a critical functional role in the overall cooling strategy. Its primary purpose is to pre-cool the incoming process air below the wet-bulb temperature, thereby significantly reducing the thermal load on the second-stage M-cycle cooler. This staged approach improves overall system efficiency and enhances the second stage's ability to achieve deeper cooling with lower air saturation deficits. While the first stage alone may not achieve the dew-point effectiveness of a regenerative M-cycle, its integration within a two-stage sequence results in synergistic performance gains, high product air ratios, and improved operating stability under diverse ambient conditions.

This study further investigates the performance of the two-stage DPEC through simulation analysis under realistic operating conditions, particularly those typical of hot summer days in subtropical climates. A three-dimensional numerical model is developed to simulate the full-scale air channels of the cooler and is validated against experimental data from the prototype. The validated model is then employed to gain deeper insights into system behavior by comprehensively comparing the two-stage configuration with conventional single-stage wet-bulb and dew-point evaporative cooling systems. Key performance indicators—such as product air temperature, product air ratio, dew-point effectiveness, cooling capacity, and cooling COP—are evaluated. Notably, these comparative insights would be difficult to obtain through experimental analysis alone.

This work offers several key contributions to the field of evaporative cooling:

- It presents the first comprehensive analysis of a two-stage DPEC system achieving a high product air ratio (~90%) and dew-point effectiveness up to 95%, surpassing state-of-the-art single-stage systems, through detailed three-dimensional simulation.
- It develops and validates a COMSOL-based numerical model capable of evaluating system performance under various design parameters and climatic conditions.
- It delivers critical insights into system design and performance optimization, informing future improvements.

Together, these contributions demonstrate the feasibility and superior performance of the proposed two-stage approach and provide a solid foundation for advancing next-generation sustainable cooling technologies.

2. Modeling methodology

In this study, the cooler was modeled using the COMSOL Multiphysics platform, a widely adopted software for simulating dew-point evaporative coolers [26,27,32]. A schematic overview of the model development and deployment workflow is presented in Figure 2.

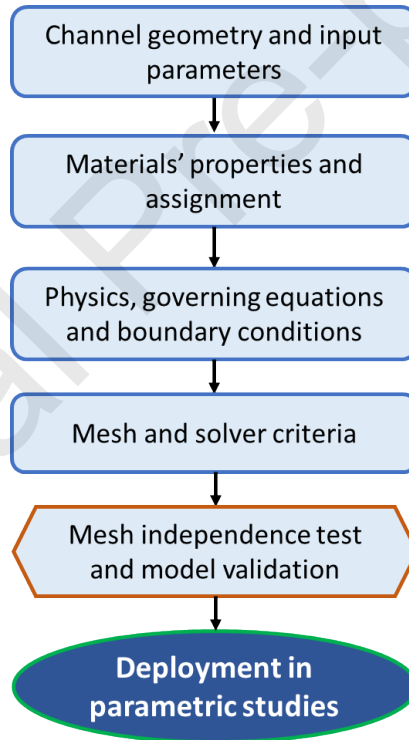


Figure 2: Schematic workflow of model development and deployment

2.1. Model geometry and assumption

The model was developed to simulate the actual three-dimensional effective channel area of the heat and mass exchanger in the reported cooler. Leveraging the channels' symmetry, the model geometry was constructed to include a half-section of an upper air channel, the channel separator with attached wet cloth layers, and a half-section of a lower air channel (Figure 3). This configuration simplifies calculations while preserving the key thermal and fluid dynamics characteristics of the system.

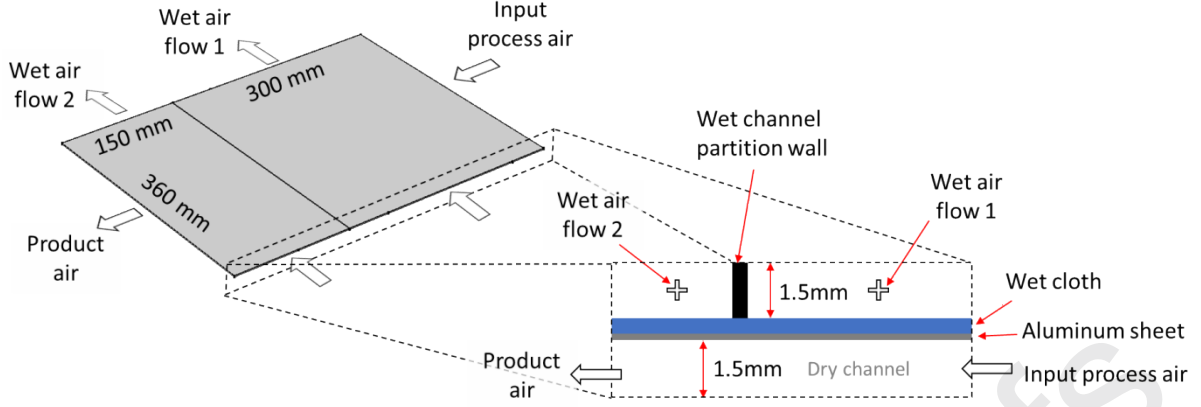


Figure 3: Geometry of the simulation model

For computational simplicity, several assumptions were made:

- The boundaries are adiabatic and sealed, meaning no heat loss or air leakage occurs between the air channels or to the external environment.
- Air is an incompressible ideal gas, following ideal gas behavior.
- Water is uniformly distributed within the wick material, ensuring complete coverage of the wet channel surfaces.

2.2. Physics, governing Equations and boundary conditions

With the operating air speed in dry and wet channels lower than 6 m/s, the corresponding Reynolds number remains below 2300. This confirms that the air flow in the channels fall in laminar regime. Therefore, the air flow is simulated using the incompressible laminar flow module in the COMSOL Multiphysics platform. The governing equations used to simulate conjugate heat and mass transfer in the air channels are as follows:

Continuity equation:

$$\rho \nabla \mathbf{u} = 0(1)$$

Momentum conservation:

$$\rho(\mathbf{u} \cdot \nabla) \mathbf{u} = \nabla \cdot [-P\mathbf{I} + \mu(\nabla \mathbf{u} + (\nabla \mathbf{u})^T)] + \mathbf{F}(2)$$

Mass transfer of water vapor:

$$M_v \mathbf{u} \cdot \nabla c_v + \nabla \mathbf{g} = (3)$$

$$\mathbf{g} = -M_v D \nabla c_v(4)$$

$$c_v = \phi c_{sat}(5)$$

Energy conservation:

$$\rho C \mathbf{u} \cdot \nabla T + \nabla \cdot \mathbf{q} = Q(6)$$

$$\mathbf{q} = -k\nabla T(7)$$

Boundary Conditions:

At the channel inlet, uniform velocity, moisture, and thermal energy conditions are imposed:

$$\mathbf{u} = -U_{in}\mathbf{n}(8)$$

$$\phi = \phi_{in}(9)$$

$$-\mathbf{n} \cdot \mathbf{q} = \rho \cdot h_{in} \cdot \mathbf{u} \cdot \mathbf{n}(10)$$

At the outlet, pressure, vapor, and heat fluxes are specified as:

$$[-P\mathbf{I} + \mathbf{K}]\mathbf{n} = -P_{amb}\mathbf{n}(11)$$

$$-\mathbf{n} \cdot D\nabla(\phi c_{sat}) = (12)$$

$$-\mathbf{n} \cdot \mathbf{q} = (13)$$

At the wet channel surfaces, evaporation leads to moisture mass flux and latent heat removal:

$$-\mathbf{n} \cdot \mathbf{g} = G_{evap}(14)$$

$$Q_{evap} = -LG_{evap}(15)$$

$$G_{evap} = M_v R(c_{sat} - c_v)(16)$$

2.3. Mesh strategy and solver criteria

The finite element method was employed to solve the coupled equations governing fluid flow, heat transfer, and mass transfer in the constructed model. The geometry was discretized into multiple computational domains using tetrahedral elements generated by COMSOL's physics-controlled mesh settings. Local mesh refinement was applied near solid–fluid interfaces, inlet/outlet boundaries, and wet surfaces to accurately capture gradients in velocity, temperature, and vapor concentration.

The simulation was performed under a stationary (steady-state) study, using a segregated solver approach for flow, thermal, and species transport modules. The PARDISO direct solver was used for linear systems, with a relative tolerance of 0.0001 set as the convergence criterion. All simulations were performed using double-precision arithmetic to ensure numerical stability.

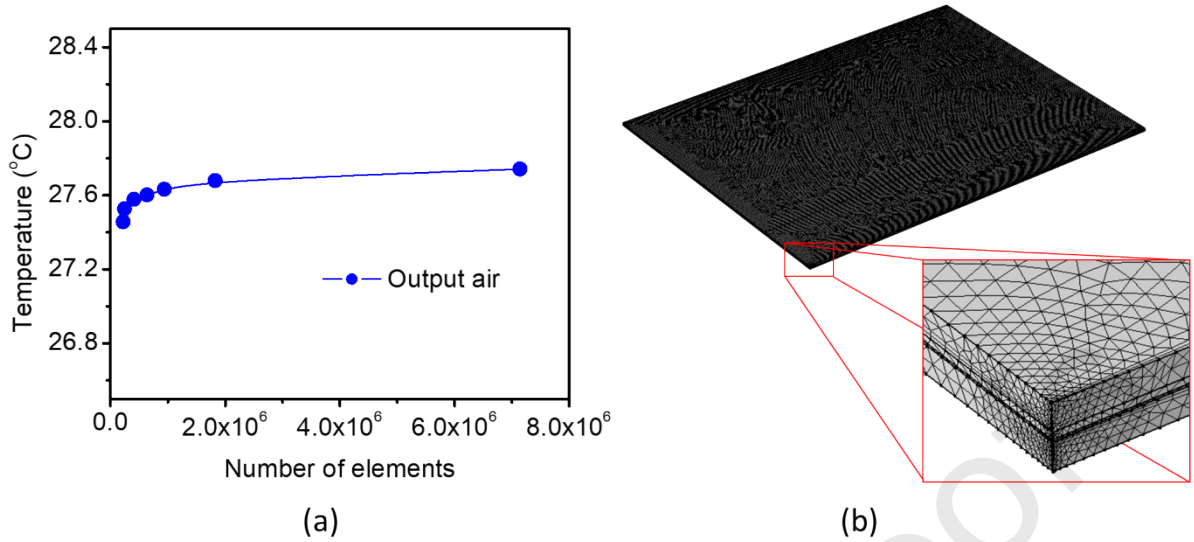


Figure 4: (a) the dependence of simulated product air temperature on the mesh size and (b) mesh scheme with 1,824,747 domains

As illustrated in Figure 4 (a), the temperature of the outlet air reached a relative stable state when the total number of discretized domains exceeded 1,000,000. Although a finer mesh enhances computational accuracy, it demands more computational resources and increases the calculation time. To optimize the trade - off between computational efficiency and accuracy, the geometry was ultimately discretized into 1,824,747 domains, as depicted in Figure 4 (b).

2.4. Model validation

Model validation was conducted using experimental performance data obtained from the developed cooler prototype operating under outdoor conditions representative of a typical sunny summer day in Zhenjiang City, China. The ambient air during testing had a dry-bulb temperature of 32.5 ± 0.5 °C and a wet-bulb temperature of 25 ± 0.5 °C.

In the experimental setup, the inlet process air flow rate was varied from 159 m³/h to 1221 m³/h. Approximately 10% of this air stream was recirculated to serve as working air for the second-stage wet channel, while the remaining 90% get out the dry channel to be product air. The total flow rate of the working air (comprising both Stage 1 and Stage 2) was maintained at 975 m³/h.

For the simulation analysis, the validated numerical model was supplied with the experimentally measured inlet air temperature, humidity, and individual air flow rates for the dry channel, wet channel 1, and wet channel 2.

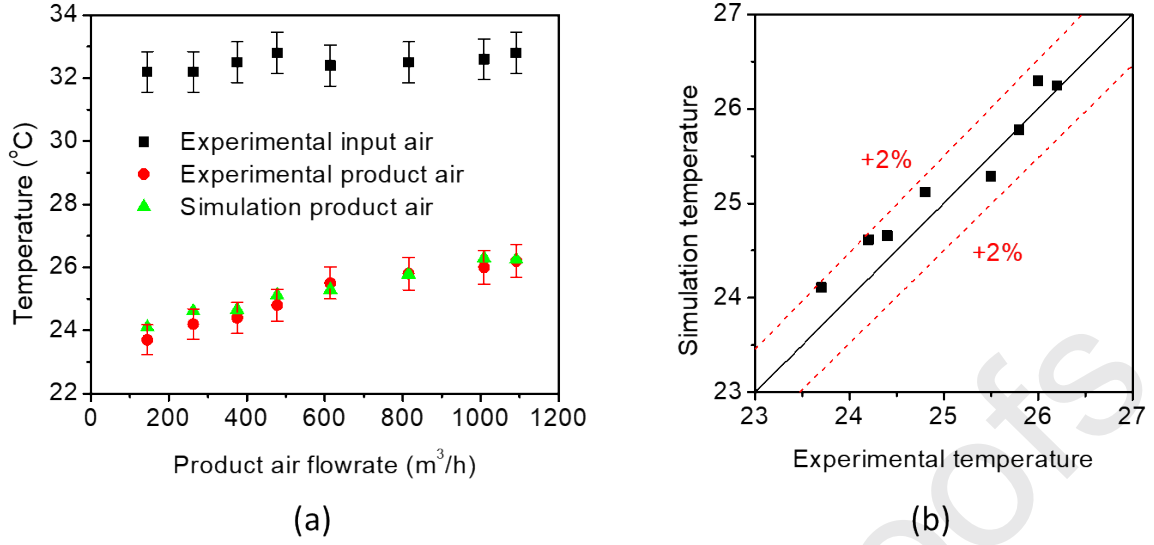


Figure 5: (a) Comparison of experimental and simulated product air temperatures as functions of product air flow rate; (b) Direct deviation of simulated product air temperature from experimental values.

Figure 5 (a) compares the simulated and experimental outlet product air temperatures as functions of product air flow rate, showing close agreement between the two datasets. Figure 5 (b) further illustrates that the discrepancies between simulated and experimental results remain within 2% across all tested conditions. This strong correlation confirms the model's reliability and predictive accuracy, validating its use for detailed parametric analyses that would be difficult to conduct experimentally.

2.5. Cooling performance evaluation

The product air ratio, r , is defined as the ratio of the product air flow rate, $\dot{V}_{in}$, to the input air flow rate, $\dot{V}_{pro}$:

$$r = \frac{\dot{V}_{pro}}{\dot{V}_{in}} \quad (17)$$

The working air ratio, $(1 - r)$, represents the fraction of product air diverted to the wet channel as working air.

The cooling performance is evaluated using wet-bulb effectiveness, ξ_{wb} , and dew-point effectiveness, ξ_{dp} . Wet-bulb effectiveness measures the temperature reduction relative to the wet-bulb cooling potential:

$$\xi_{wb} = \frac{T_{in} - T_{pro}}{T_{in} - T_{wb}} \times (18a)$$

Similarly, dew-point effectiveness measures the temperature reduction relative to the dew-

point cooling potential of the input air:

$$\xi_{dp} = \frac{T_{in} - T_{pro}}{T_{in} - T_{dp}} \times (18b)$$

The cooling capacity, H , is calculated from the mass flowrate of the product air, as:

$$H = (T_{in} - T_{pro}) \times \dot{m}_{pro} \times 1.006 \frac{kJ}{kg \cdot K} (19)$$

The blower power, E , depends on the air flow rate, $\dot{V}$, the static pressures drop, ΔP , and fan efficiency, η . The fan efficiency $\eta = 0.36$ is fixed for all calculations. Blower power is the primary energy consumption; thus, selecting a high-efficiency fan is critical.

$$E = \frac{\Delta P \cdot \dot{V}}{\eta} (20)$$

The static pressure drop varies with the airflow rate, $\dot{V}$, in each channel. This relationship is derived from experimental measurements and is presented in the revised Figure 6.

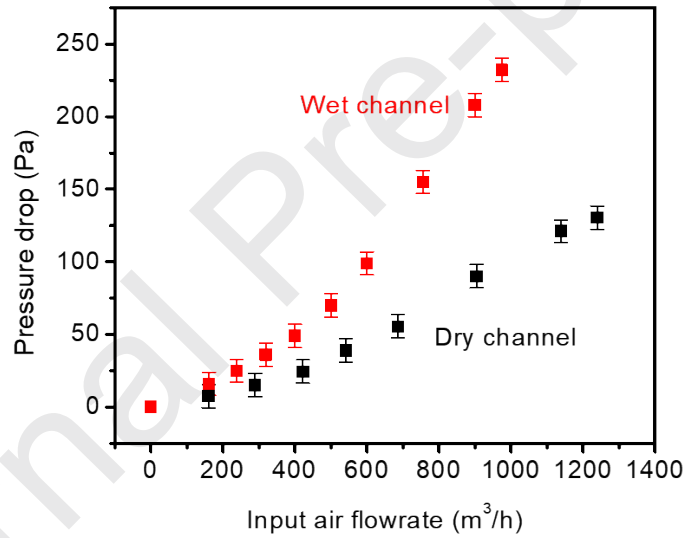


Figure 6: Experimental relationship between pressure drops and air flowrates in the wet and dry channels.

The COP is the ratio of cooling capacity, H , to blower power, E .

$$COP = \frac{H}{E} (21)$$

3. Results and discussion

3.1. Psychrometric cooling processes in the developed system

To elucidate the cooling mechanisms within the cooler, the simulation model was used to

generate temperature and humidity ratio distributions across the wet and dry channels under ambient conditions of 32.30°C and 70% relative humidity. As shown in Figure 7 (a), the process air inlet flowrate was set at 500 m³/h. And in Figure 7 (b), the first cooling stage employs 200 m³/h of ambient air as working fluid, while the second stage diverts 100 m³/h of product air from the dry channel outlet to the wet channel inlets.

Figure 7 (a) illustrates that the process air undergoes sensible cooling (constant moisture content) from ambient temperature to 28.12°C at the end of the first stage, followed by further cooling to 26.98°C at the dry channel outlet. This stepwise cooling reflects the dual-stage design. Conversely, the wet channel experiences simultaneous temperature reduction and humidity increase due to evaporative cooling in both stages (Figure 7 (b) and 7 (c)). The evaporation rate peaks near the heat exchanger entrance, where the humidity gradient is highest, and decreases along the airflow path. This aligns with the observed temperature profiles in both channels. The working air exits both stages approaching 100% relative humidity, indicating near-saturation conditions.

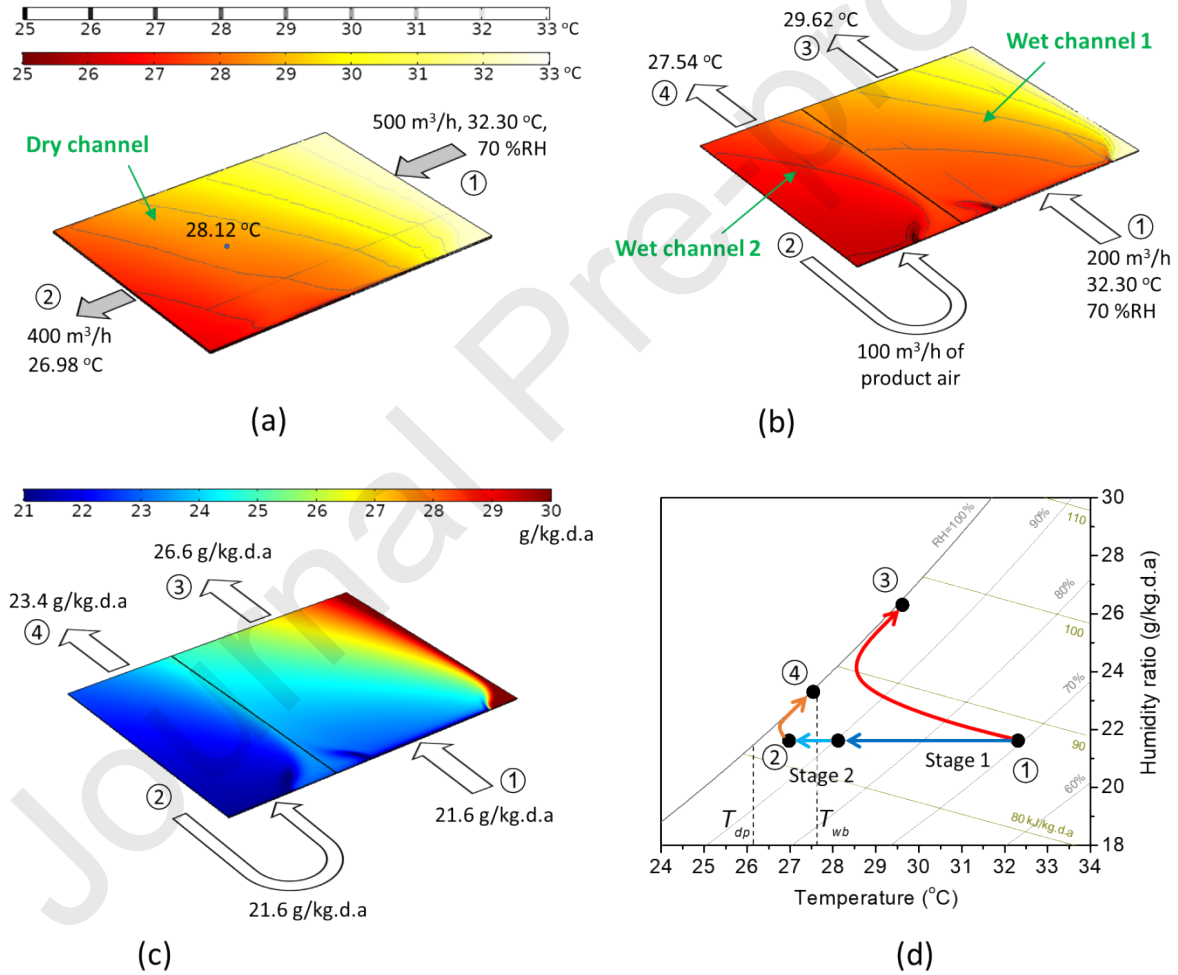


Figure 7: (a) air temperature distribution in dry channel, (b) air temperature distribution in wet channels, (c) humidity ratio distribution in wet channels, and (d) the psychrometric change in the air conditions during the cooling process in the developed two-stage dew-point evaporative cooler

Figure 7 (d) depicts the psychrometric transformations. In the first stage, ambient air as working fluid cools the process air toward the wet-bulb temperature. In the second stage, recirculated product air as working fluid enables cooling below the wet-bulb temperature, approaching the dew-point. This multi-stage approach enhances cooling efficiency beyond conventional evaporative systems by leveraging internal heat recovery and staged evaporation.

3.2. Effect of wet-channel width on cooling performance

An important consideration in the two-stage dew-point cooling system is the impact of wet-channel width on the cooler's performance. Experimental analysis of this factor is impractical due to the extensive fabrication efforts and costs involved. However, the developed simulation model provides an efficient means to investigate this effect.

The simulation is performed by utilizing the air flowrates in the dry and wet channels, as illustrated in Figure 8 (a). During the simulation, the width of wet channel 2 is varied from 5 cm to 20 cm, which is achieved by moving the partition wall between the wet channels. This variation corresponds to a reduction in the width of wet channel 1 from 40 cm to 25 cm. When the working air flowrates are held constant, an increase in the wet-channel width of a channel results in a decrease in its working air velocity, and conversely, a decrease in wet-channel width leads to an increase in working air.

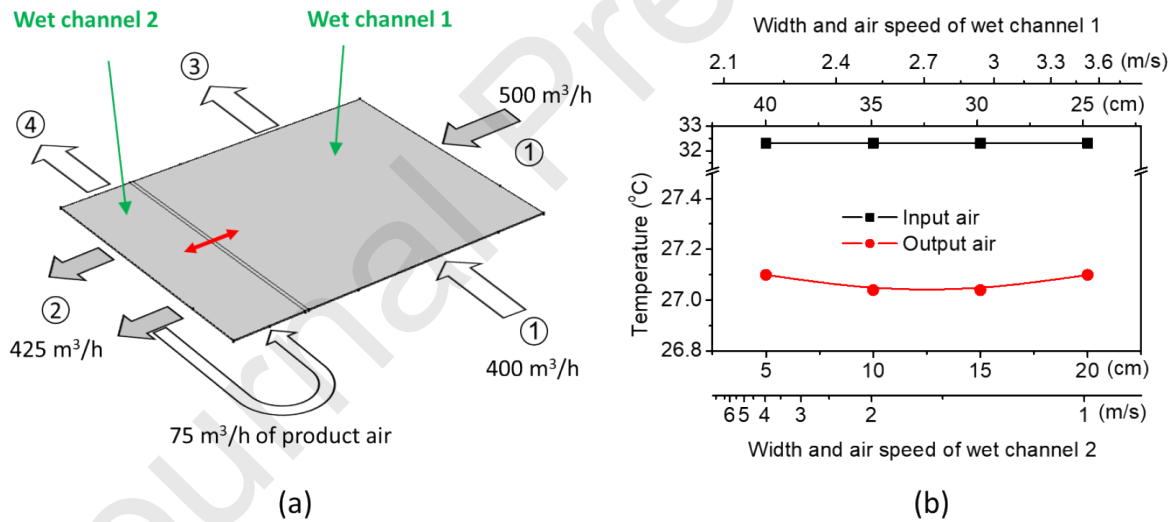


Figure 8: (a) air volume flowrates in dry and wet channels and (b) change in the product air temperature with the variation of the widths of the wet channel cooling stages.

Figure 8 (b) illustrates the product air temperature as a function of the width of wet channel 2. The results indicate that variations in wet-channel width have minimal impact on cooling performance. The product air temperature remains relatively stable, reaching its lowest value when wet-channel 2 width is between 10 cm and 15 cm. This stability is attributed to the current channel dimensions, specifically the 3 mm height and 36 cm length, which effectively limit bypass effects across the tested airflow conditions. Instead, cooling performance is more

significantly influenced by the working air volume flowrates, which will be explored further in subsequent sections.

3.3. Performance comparison: M-cycle mode vs. two-stage dew-point cooling mode

To evaluate the performance of the developed two-stage dew-point cooler relative to a conventional M-cycle cooler, simulations are conducted in two modes. Figure 9 (a) illustrates the M-cycle cooling mode, where the working air for both types of wet channels is sourced solely from the product air. In contrast, Figure 9 (b) presents the two-stage cooling mode, in which the working air for the first cooling stage is drawn from ambient air, while the working air for the second cooling stage is derived from the product air.

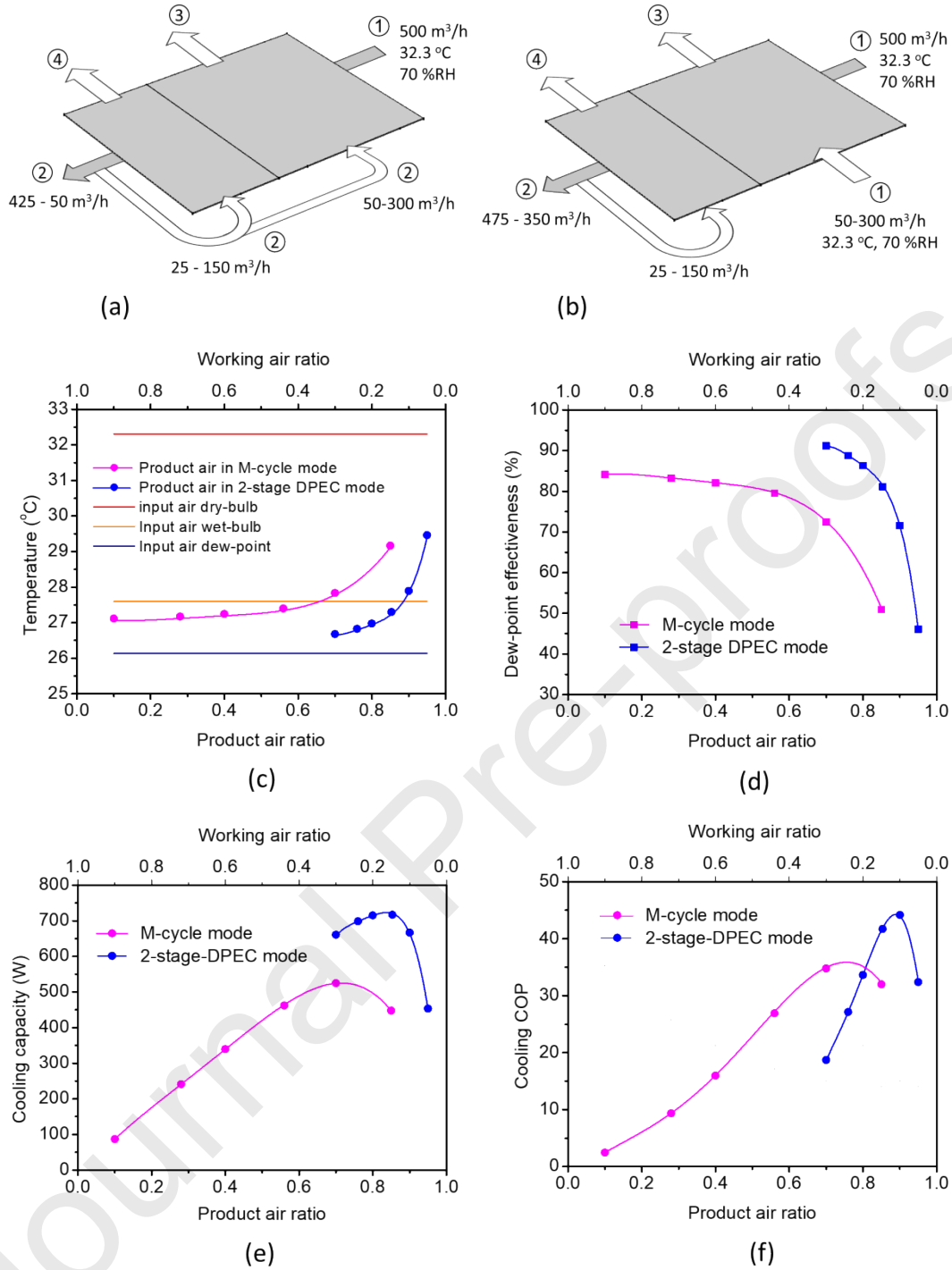


Figure 9: Air flow conditions in (a) M-cycle cooling mode and (b) two-stage dew-point cooling mode; (c) product air temperature profile, (d) dew-point effectiveness, (e) cooling capacity, and (f) cooling COP at different product air ratio of the two cooling modes.

To ensure a fair comparison, all operational parameters remain identical across both simulations. The process air flowrate is set at 500 m³/h. The working air flowrate in wet channel 2 is varied from 25 m³/h to 150 m³/h, while the working air flowrate in wet channel 1 is twice this value, ranging from 50 m³/h to 300 m³/h, proportional to their respective channel widths. The usable product air flowrate in the M-cycle mode varies from 425 m³/h to 50 m³/h, which

is lower than that observed in the two-stage cooling mode, where it ranges from 475 m³/h to 350 m³/h. The input air conditions correspond to typical tropical ambient air parameters.

The product air temperature profiles for both cooling modes are shown in Figure 9 (c). The results indicate that cooling performance improves at higher working air ratio, corresponding to lower product air flowrates, as the returning product air effectively cools the process air below the wet-bulb temperature, reaching the dew-point temperature in both modes. However, Figure 9 (d) highlights a notable advantage of the two-stage cooling mode, which achieves deeper cooling and higher dew-point effectiveness at a higher product air ratio. This enhanced performance underscores the benefits of incorporating ambient air in the first cooling stage, making the two-stage dew-point cooler a more efficient alternative to conventional M-cycle systems.

The cooling capacities of the two simulated modes, derived from the product air ratios and temperature reductions following Equation (19), are presented in Figure 9 (e). The results indicate that cooling capacities decrease as the product air ratio approaches its limits at 0 and 1, where either the product air flowrate or the temperature reduction diminished. Each cooling mode exhibits an optimal product air ratio at which its cooling capacity peaks. Under the tested conditions, the M-cycle mode reaches its peak cooling capacity at a product air ratio of 0.7, while the two-stage cooling mode achieves its highest efficiency at 0.85. Due to its ability to retain more product air, the two-stage cooling mode demonstrates superior performance, with a peak cooling capacity 36% higher than that of the M-cycle mode.

It is important to note that the air flowrates in the wet and dry channels remain identical for both cooling modes, ensuring that their fan power consumption—calculated based on pressure drop, air volume flowrate, and motor efficiency following Equation (20)—is equivalent. The cooling COP for each mode is then determined as the ratio of cooling capacity to total fan power consumption, as shown in Figure 9 (f). Similar to cooling capacity, there exists an optimal product air ratio at which COP reaches its maximum. The peak COP values for the M-cycle and two-stage cooling modes occur at approximately 0.75 and 0.9, respectively. At these peak values, the two-stage cooling mode demonstrates 25% higher energy efficiency compared to the M-cycle mode, highlighting its effectiveness in achieving superior cooling performance with optimized energy use.

3.4. Performance comparison: IEC mode vs. two-stage DPEC mode

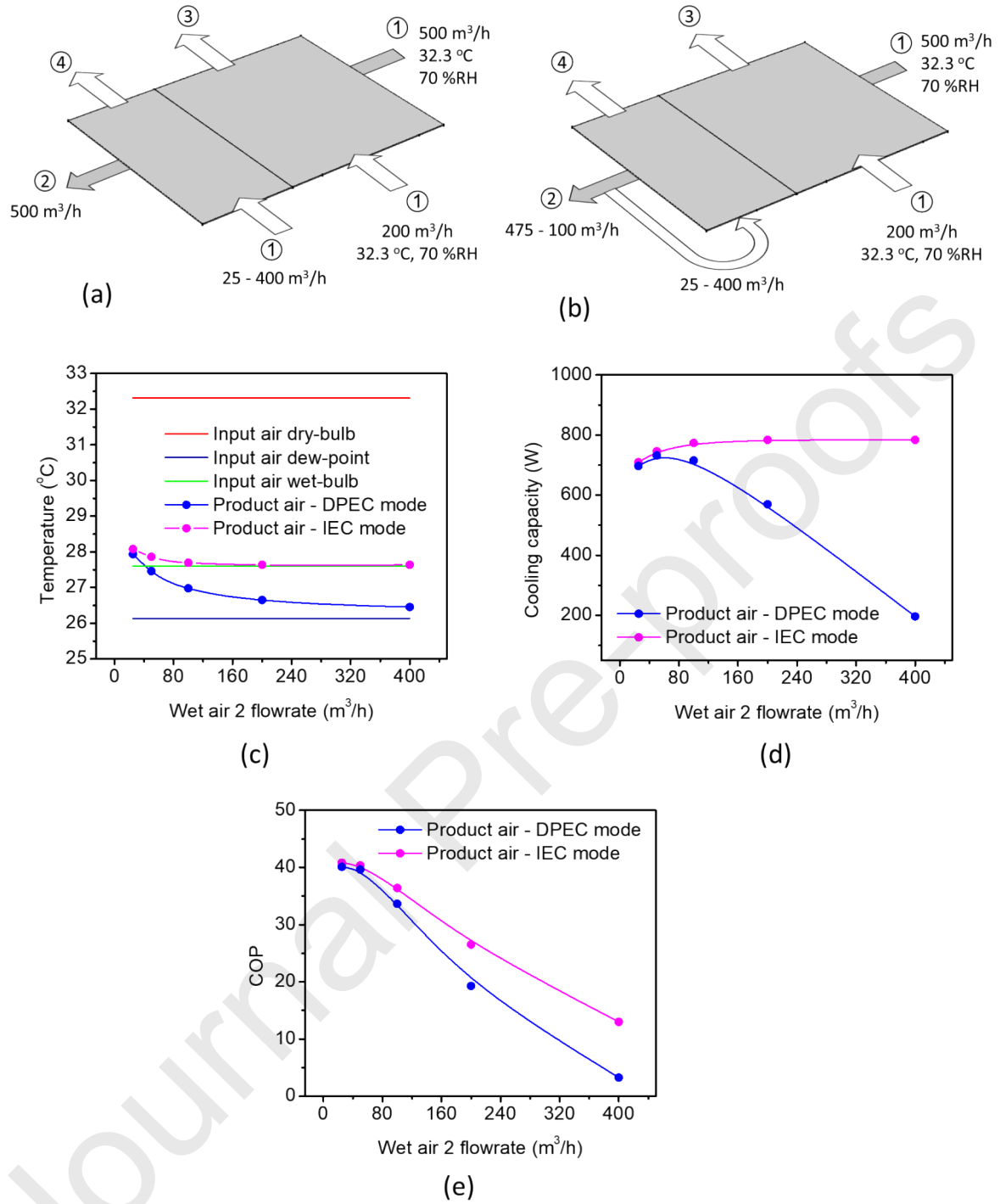


Figure 10: Air flow conditions in (a) conventional IEC mode and (b) two-stage DPEC mode; (c) product air temperature, (d) cooling capacity and (e) cooling COP at different wet air 2 flowrate in the two cooling modes.

The advantages of the developed two-stage DPEC over the conventional IEC system are evaluated through simulation analyses. The model is run in two configurations: the IEC mode, where the working air for the wet channels is sourced solely from ambient air (Figure 10 (a)), and the two-stage DPEC mode, where product air is utilized as the working air in the second cooling stage (Figure 10 (b)). For an objective comparison, identical input air flowrates are maintained in both modes. The dry channel and wet channel 1 are supplied with ambient air at

flowrates of 500 m³/h and 200 m³/h, respectively, while the working air flowrate in wet channel 2 is varied from 25 m³/h to 400 m³/h.

The product air temperature profiles for both cooling modes are presented in Figure 10 (c). The two-stage DPEC mode demonstrates superior cooling performance, effectively lowering the product air temperature below the wet-bulb temperature and approaching the dew-point temperature, whereas the IEC mode is constrained to the wet-bulb temperature limit. This underscores the importance of pre-cooled working air in achieving deeper cooling in the second stage. However, this temperature reduction comes at the expense of usable product air flowrate. Consequently, the cooling capacity of the two-stage DPEC mode is lower than that of the IEC mode, as depicted in Figure 10 (d). Beyond the peak cooling capacity, the two-stage DPEC mode exhibits a sharp decline when wet air 2 flowrate increases, whereas the cooling capacity of the IEC mode stabilizes as its product air temperature approaches the wet-bulb temperature at higher wet air 2 flowrates.

Considering fan power consumption—which continues to rise with increasing wet air 2 flowrates—the cooling COP for both cooling modes declines under higher wet air flow conditions (Figure 10 (e)). However, an optimal balance is observed when the wet air 2 flowrate is maintained between 50 m³/h and 100 m³/h, corresponding to 10-20% of product air being used as working air. This range effectively balances significant cooling improvements with acceptable energy efficiency, making the two-stage DPEC mode a viable enhancement over the conventional IEC system.

3.5. Effect of working air flowrates on cooling performance

Figure 8 demonstrates that variations in the width of the wet channels have minimal impact on the cooling performance of the developed two-stage DPEC system. Instead, the working air flowrates in the two wet channels play a more significant role, as illustrated in Figure 11. In this analysis, the process air flowrate in the dry channels is maintained at 500 m³/h. The effects of varying the working air flowrates in wet channels 1 and 2—ranging from 50 to 400 m³/h and 25 to 100 m³/h, respectively—on product air temperature are depicted in Figure 11 (a).

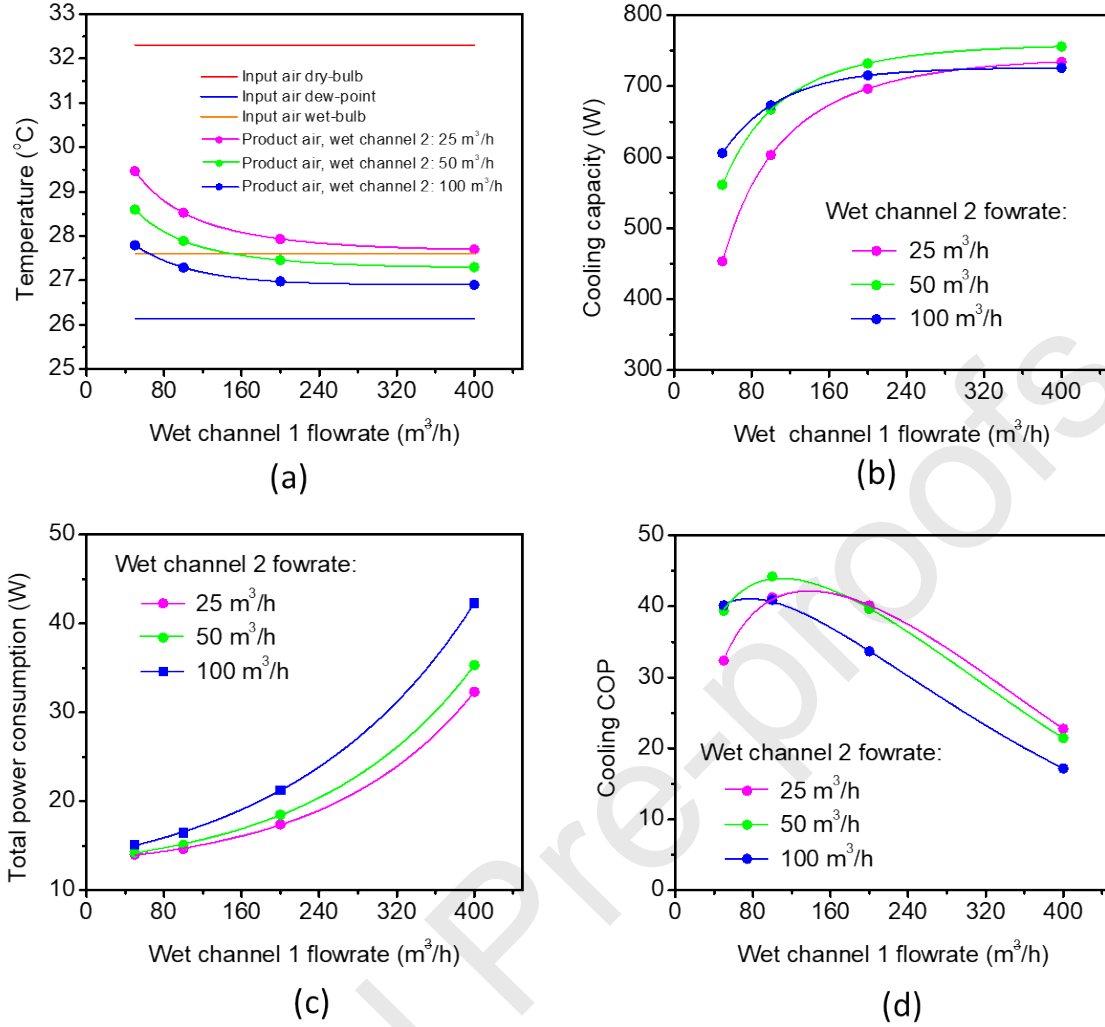


Figure 11: Changes in (a) product temperatures, (b) cooling capacity, (c) total power consumption and (d) cooling COP of the developed two-stage dew-point evaporative cooler at different working air flowrates.

The results indicate that increasing working air flowrates enhances product air temperature reduction. To achieve a product air temperature lower than the wet-bulb temperature, at least 25 m³/h of product air (equivalent to 5% of total product air) should be used as working air in the second cooling stage. Additionally, increasing the working air flowrate in wet channel 1 results in further cooling. However, beyond 200 m³/h, the product air temperature and cooling capacity (Figure 11 (b)) stabilize, providing little additional benefit. Moreover, excessive working air flowrates lead to a rapid increase in fan power consumption (Figure 11 (c)), subsequently reducing the cooling COP, as shown in Figure 11 (d).

Considering the trade-off between product air temperature reduction and energy efficiency, optimal cooling performance is achieved when 50-100 m³/h (10-20% of total product air) is diverted as working air in wet channel 2, while 100-200 m³/h of ambient air is supplied to wet channel 1. This range provides an effective balance between cooling efficiency and energy consumption, ensuring reliable performance of the two-stage DPEC system.

3.6. Effect of input air temperature and humidity ratio on cooling performance

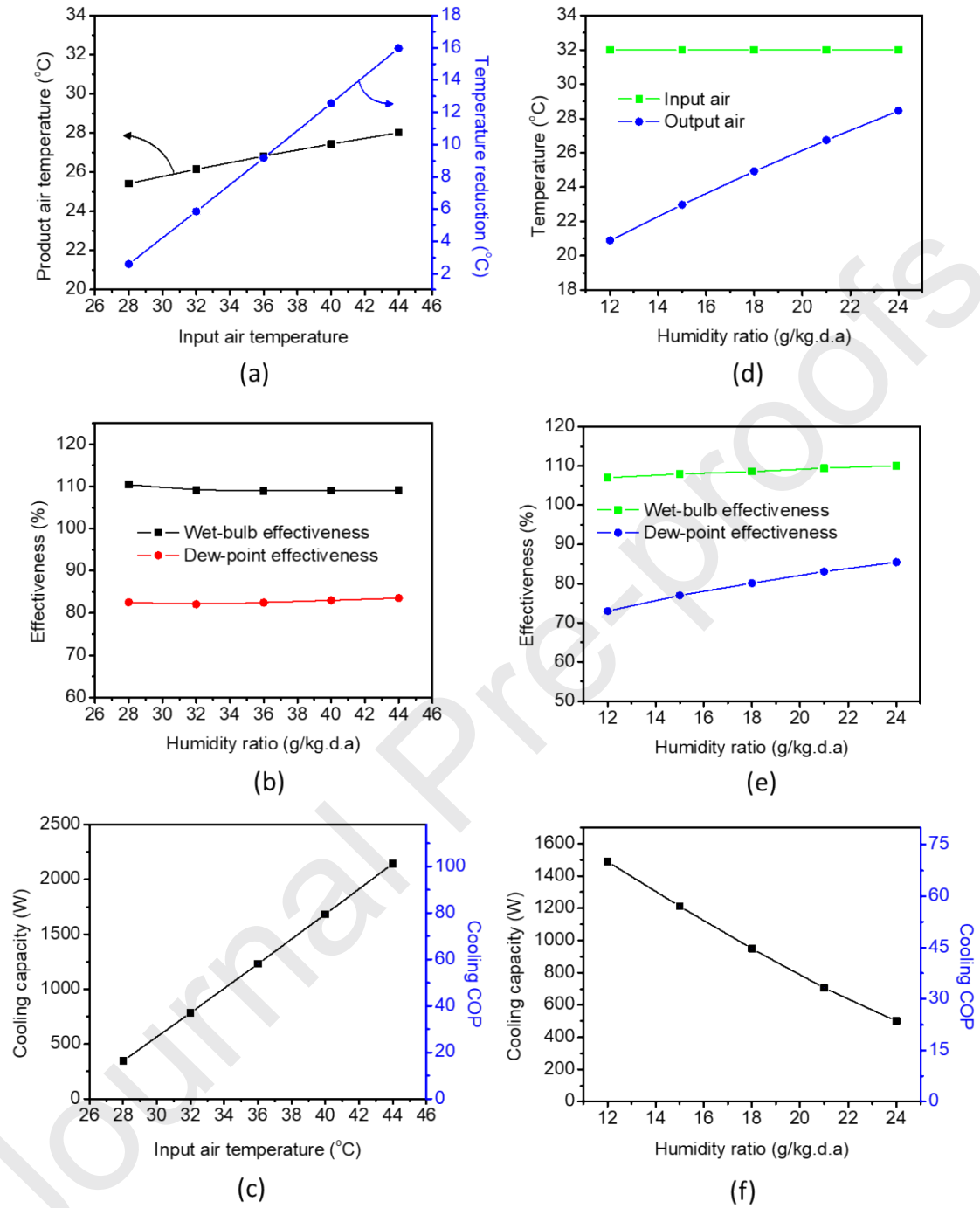


Figure 12: (a) product air temperature and temperature reduction, (b) dew-point and wet-bulb effectivenesses, and (c) cooling capacity and COP as functions of input air temperature; (d) product air temperature, (e) dew-point and wet-bulb effectivenesses, and (f) cooling capacity and COP as functions of input humidity ratio.

The effects of input air temperature and humidity conditions on the cooling performance of the two-stage DPEC system are analyzed under optimal flowrate conditions, where the process air is set at 500 m³/h, working air 1 at 200 m³/h, and working air 2 at 75 m³/h, corresponding to a product air ratio of 85%.

In the first set of tests, the input air temperature is varied from 28°C to 44°C while maintaining a constant input air humidity ratio of 20 g/kg.d.a. Figure 12 (a) illustrates the relationship between the product air temperature and the input air temperature. The results indicate a marginal increase in the product air temperature and rapid increase in the temperature reduction with higher input air temperatures. Figure 12 (b) shows that the dew-point and wet-bulb effectivenesses remain stable at approximately 83% and 108%, respectively. Increased input air temperature enhances cooling performance, leading to greater temperature reduction (Figure 12 (a)) and higher cooling capacity (Figure 12 (c)). Variations in input air temperature do not impact total fan power consumption. Consequently, cooling COP, which is proportional to cooling capacity, increases linearly with higher input air temperatures (Figure 12 (c)).

In the second set of tests, the ambient humidity ratio is varied from 12 to 24 g/kg.d.a while the air temperature is maintained at 32°C. Figure 12 (d) demonstrates that product air temperature rises as the air humidity ratio increases. This effect is attributed to the thermodynamic limitation imposed by the input air dew-point temperature, which increases with higher humidity levels. Under the tested conditions, the cooler's dew-point effectiveness remains stable at approximately 108%, as shown in Figure 12 (e). However, higher input air humidity ratios lead to reduced cooling performance, evidenced by a linear decline in cooling capacity and a decrease in cooling COP (Figure 12 (f)).

Overall, the results confirm that the two-stage DPEC system performs optimally in hot and dry conditions. Its cooling capacity reaches up to 2.1 kW, with COP values ranging from 16 to 100, depending on the input air conditions.

3.7. Cooling performance across different climatic conditions

To assess the effectiveness of the developed dew-point cooler in various climatic conditions worldwide, a simulation analysis was conducted using input air parameters corresponding to the average daytime temperature and relative humidity during the three hottest months in cities with peak cooling demand. The selected locations—Athens, Doha, Las Vegas, Madrid, and Mumbai—represent Mediterranean, hot desert, cold desert, semi-arid, and tropical monsoon climates, respectively (Table 1).

Table 1: Typical climates types, example cities, and their average daytime weather conditions in three hottest months in 2024 obtained from <https://www.visualcrossing.com/>

Climates	Cities	Three hottest months	Average daytime temperature (°C)	Average daytime relative humidity (%)
Mediterranean	Athens	June - August	32.44	30.58
Hot desert	Doha	June - August	38.80	40.16

Cold desert	Las Vegas	June - August	38.98	11.76
Semi-arid	Madrid	June - August	28.82	34.12
Tropical monsoon	Mumbai	April - June	31.77	66.06

Three operational configurations were examined: conventional IEC mode, M-cycle mode, and the two-stage DPEC mode, as depicted in Figure 13 (a)–13 (c). To ensure a fair comparison, identical air flowrates were maintained across all three modes: 240 m³/h in wet channel 1 and 80 m³/h in wet channel 2. The resulting product air temperature, wet-bulb effectiveness, cooling capacity, and COP for the three cooling configurations are presented in Figure 13 (d)–13 (f).

Figure 13 (d) illustrates that, under the analyzed airflow conditions, the IEC mode exhibits the highest product air temperature among the three configurations, and higher than the wet-bulb temperature. Consequently, the wet-bulb effectiveness in IEC mode remains below 100%, as shown in Figure 13 (e). In contrast, both the M-cycle and two-stage DPEC modes achieve cooling below the wet-bulb temperature, resulting in the wet-bulb effectiveness values exceeding 100%. Between these two modes, the M-cycle configuration yields slightly lower product air temperatures than the two-stage DPEC mode.

For both the M-cycle and two-stage DPEC modes, product air temperatures of 25–26°C are attainable in humid climates such as Doha and Mumbai, whereas significantly lower temperatures of 15–20°C are observed in arid and semi-arid environments such as Athens, Las Vegas, and Madrid. These findings underscore the adaptability of the cooler as either a standalone solution or a pre-cooling system for conventional air conditioning setups. Moreover, temperature reduction is more pronounced in dry air conditions, while humid air environments limit the cooling effect, reflecting the inherent thermodynamic relationship between ambient moisture content and evaporative cooling efficiency.

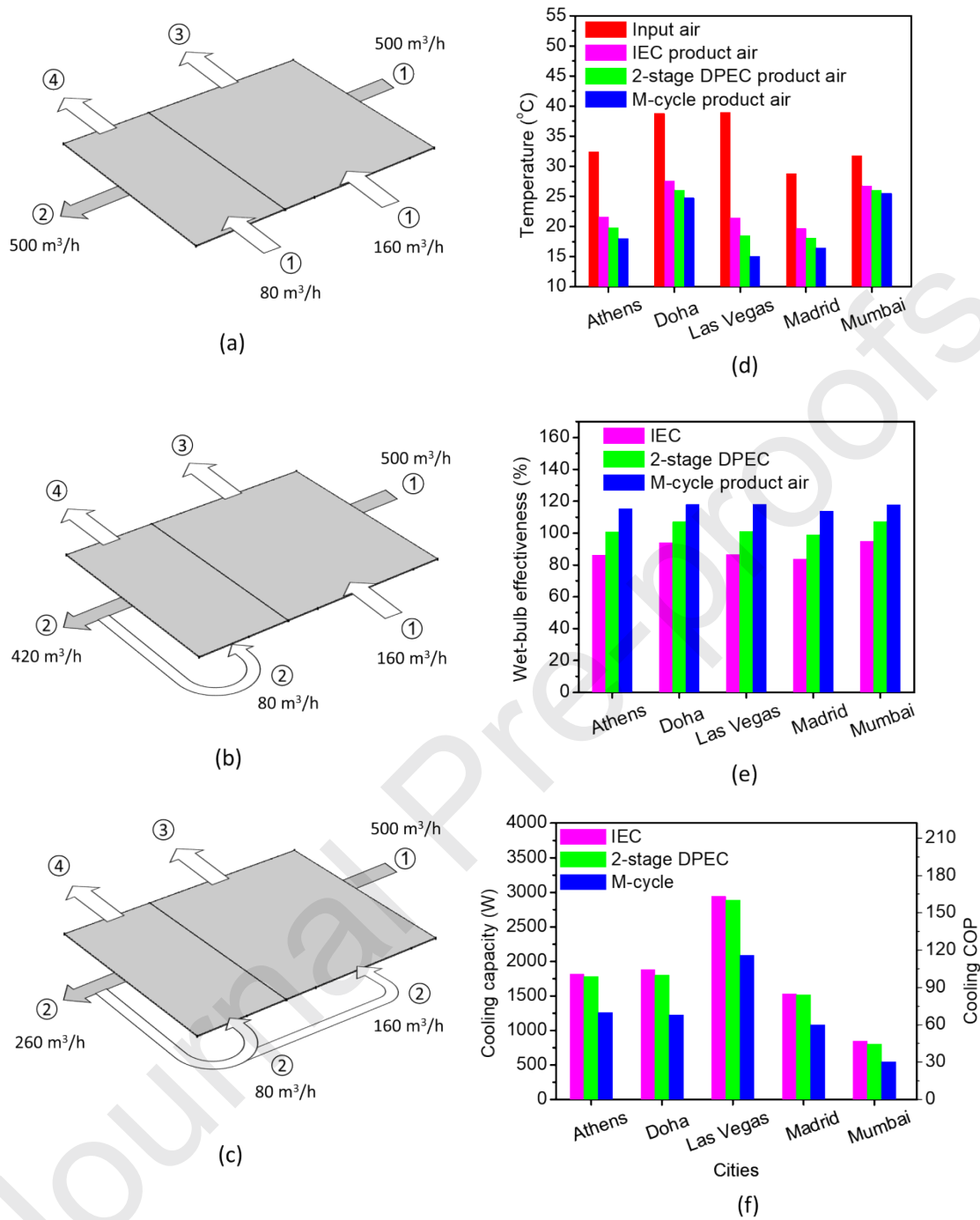


Figure 13. Performance comparison of the two-stage DPEC, M-cycle, and conventional IEC under typical daytime weather conditions during the three hottest summer months across different cities: (a) airflow rates in IEC mode, (b) two-stage DPEC modes, and (c) M-cycle mode; (d) input and output air temperatures; (e) wet-bulb effectiveness; and (f) average cooling capacity and COP across different modes and cities.

Figures 13 (d) and 13 (e) indicate that increasing the amount of pre-cooled product air used as working air enhances the temperature reduction and wet-bulb effectiveness. However, this approach reduces the usable product air flowrate. In the M-cycle mode, a largest proportion of

product air is utilized as working air, resulting in the lowest usable product air flowrate. In contrast, the IEC mode, which does not utilize product air for working air, maintains the highest usable product air flowrate. Consequently, the M-cycle mode exhibits the lowest cooling capacity and COP, while the IEC mode achieves the highest values, as depicted in Figure 13 (f).

The two-stage DPEC configuration provides an optimal balance between temperature reduction and product air preservation, delivering acceptable product air temperatures, cooling capacity, and COP. The cooler demonstrates peak energy efficiency under cold desert conditions, achieving a cooling capacity of 2887 W and a COP of 160 in Las Vegas. Conversely, its efficiency is lowest in tropical monsoon climates, where the cooling capacity and COP decline to 804 W and 45 in Mumbai, respectively.

4. Conclusion

This study presents a comprehensive numerical investigation of a novel two-stage dew-point evaporative cooling (DPEC) system, designed to overcome key limitations of conventional single-stage configurations. A validated three-dimensional COMSOL model was developed to simulate the heat and mass transfer within a compact exchanger that handles four fluid streams, enabling high-resolution performance analysis under various operating and climatic conditions.

By separating the cooling process into a first-stage wet-bulb cooling using ambient air and a second-stage dew-point cooling using a fraction of the cooled product air, the system achieves deeper cooling while significantly reducing working air requirements.

Key findings from the simulation-based analysis include:

- **Enhanced Energy Efficiency:** Compared to single-stage DPECs, the two-stage configuration offers over 30% improvement in energy efficiency and cooling capacity while reducing the required working air ratio by half.
- **Optimized Operation:** Analyses indicate that optimal cooling performance occurs when 50–100 m³/h (10–20% of the total product air) is diverted as working air in the second cooling stage, with 100–200 m³/h of ambient air supplied to the first stage. The system excels in hot and dry environments, offering a cooling capacity of up to 2.1 kW and a COP ranging from 16 to 100, depending on the input air conditions.
- **Adaptability Across Climates:** Performance evaluations in diverse global climates, including Mediterranean, hot desert, cold desert, semi-arid, and tropical monsoon regions, reveal that the cooler can function effectively as a standalone solution or a pre-cooling system. Notably, it achieves peak energy efficiency in cold desert conditions, such as in Las Vegas, where a COP of 160 was recorded.

The proposed two-stage DPEC system represents a promising advancement in sustainable air conditioning technology. Its compact form factor, high energy efficiency, and reduced dependence on electrical power make it well-suited for regions facing rising cooling demand and energy constraints. By shifting away from compressor-based systems, this technology supports broader global goals for energy conservation and decarbonization.

Building on these findings, future research will explore enhancements to system architecture, such as the integration of counter-flow or quasi-counter-flow channel configurations, to improve transfer effectiveness. Strategies for managing the resulting pressure drops through optimized channel design and flow control will also be investigated. Additionally, long-term durability assessments and techno-economic analyses will support the transition of this technology toward real-world deployment.

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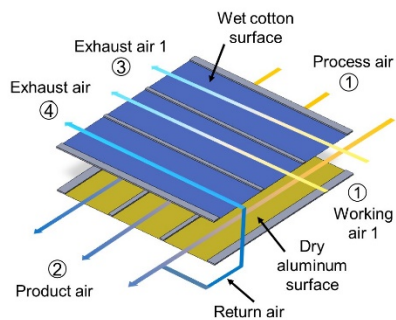
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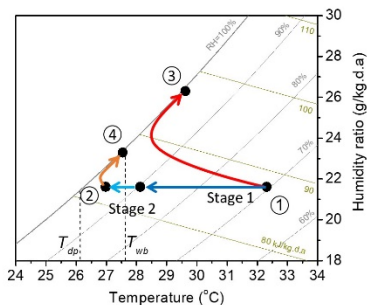
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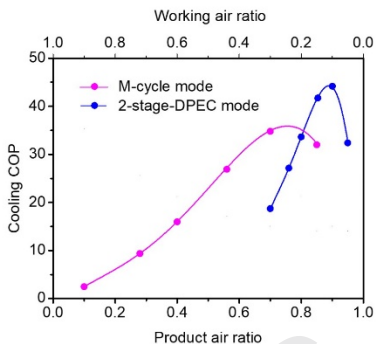
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Innovative compact four-fluid, two-stage dew-point evaporative heat and mass exchanger



Efficient cooling load distribution across two sequential evaporative stages



Enhanced cooling efficiency achieved with over 90% product air utilization

Highlights:

- A novel two-stage dew-point evaporative cooler is experimentally developed.
- A comprehensive study enables in-depth understanding and performance optimization.
- Product air ratio up to 90%, reducing working air demand and losses.
- COP reaches up to 100 under ideal conditions, outperforming single-stage systems.
- The cooler supports energy-efficient, climate-resilient cooling with global relevance.

Declaration of interests

☐ 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.

☒ The authors declare the following financial interests/personal relationships which may be considered as potential competing interests:

DT. Bui reports financial support was provided by Jiangsu University. If there are other authors, they declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.



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Energy-efficient cooling beyond M-cycle: development and evaluation of a two-stage dew-point evaporative cooler

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