Understanding the transient behavior of the dew point evaporative cooler from the first and second law of thermodynamics

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Abstract

Owing to its high energy efficiency without using greenhouse gases, dew point evaporative cooling offers a desired solution for thermal management of electronic and electrical devices. This paper elucidates the transient behavior of a dew point evaporative cooler and its significant influence on the dynamic cooling performance. A large time constant (400 s) of the product air temperature was observed under a zero-state response, leading to a pronounced deviation of the time-average cooling performance below its steady state by 13.8%–26.4% over a long period (2500 s). To capture this phenomenon, a modified transient lumped parameter model and a new partial differential exergy model were developed. An air mixing process in the dry channel was identified to account for the slow cooler’s transient responses. A detailed exergy analysis revealed that the specific exergy destruction at the dry channel entrance was above 400 W/kg, owing to the air mixing. This finding demonstrates that the transient behavior should be judiciously considered in the cooler design and optimization, together with the steady-state performance. Accordingly, a detailed sensitivity analysis of the cooler’s objective variables is proposed to gain insights into the future improvement of the dew point evaporative cooler.

Keywords: dew point evaporative cooling; transient; lumped parameter model; exergy analysis; sensitivity analysis
### Nomenclatures

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Subscript</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>(a)</td>
<td>coefficient</td>
<td>(\varepsilon)</td>
<td>effectiveness</td>
</tr>
<tr>
<td>(C)</td>
<td>coefficient</td>
<td>(\phi)</td>
<td>relative humidity</td>
</tr>
<tr>
<td>(c_p)</td>
<td>specific heat at constant pressure, J/(kg·K)</td>
<td>(\rho)</td>
<td>density, kg/m³</td>
</tr>
<tr>
<td>(c_v)</td>
<td>specific heat at constant volume, J/(kg·K)</td>
<td>(\omega)</td>
<td>humidity ratio, kg/kg</td>
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<tr>
<td>COP</td>
<td>coefficient of performance</td>
<td></td>
<td></td>
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<tr>
<td>(D_{va})</td>
<td>diffusion coefficient, m²/s</td>
<td></td>
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<tr>
<td>(E)</td>
<td>energy, J</td>
<td>(0)</td>
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<td>error</td>
<td>(a)</td>
<td>air</td>
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<tr>
<td>(ex)</td>
<td>specific exergy, J/kg</td>
<td>(CV)</td>
<td>control volume</td>
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<tr>
<td>(EX)</td>
<td>exergy, J</td>
<td>(d)</td>
<td>dry</td>
</tr>
<tr>
<td>(h)</td>
<td>specific enthalpy, J/kg</td>
<td>(db)</td>
<td>dry bulb</td>
</tr>
<tr>
<td>(\overline{h})</td>
<td>convective heat transfer coefficient, W/(m²·K)</td>
<td>(de)</td>
<td>destruction</td>
</tr>
<tr>
<td>(\overline{h}_m)</td>
<td>convective mass transfer coefficient, m/s</td>
<td>(dp)</td>
<td>dew point</td>
</tr>
<tr>
<td>(H)</td>
<td>channel height, m</td>
<td>(e)</td>
<td>energy</td>
</tr>
<tr>
<td>(J)</td>
<td>Jacobian matrix</td>
<td>(f)</td>
<td>water film</td>
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<tr>
<td>(k)</td>
<td>thermal conductivity, W/(m·K)</td>
<td>(in)</td>
<td>inlet</td>
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<tr>
<td>(L)</td>
<td>channel length, m</td>
<td>(l)</td>
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</tr>
<tr>
<td>(\dot{m})</td>
<td>mass flow rate, kg/s</td>
<td>(m)</td>
<td>mass</td>
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<tr>
<td>(M)</td>
<td>mass, kg</td>
<td>(n)</td>
<td>mass transfer</td>
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<tr>
<td>(n)</td>
<td>mass transfer rate, kg/s</td>
<td>(out)</td>
<td>outlet</td>
</tr>
<tr>
<td>(P)</td>
<td>pressure, Pa</td>
<td>(p)</td>
<td>product air</td>
</tr>
<tr>
<td>(q)</td>
<td>heat transfer rate, W</td>
<td>(q)</td>
<td>heat transfer</td>
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<tr>
<td>(\dot{Q})</td>
<td>cooling capacity, W</td>
<td>(r)</td>
<td>right</td>
</tr>
<tr>
<td>(r)</td>
<td>working air ratio</td>
<td>(s)</td>
<td>supply air</td>
</tr>
</tbody>
</table>
$R$ specific gas constant, J/(kg·K) ss steady state
$s$ specific entropy, J/(kg·K) sa saturation
$t$ time, s v water vapor
$t_c$ time constant, s w wet/working air
$T$ temperature, K wb wet bulb
$u$ specific internal energy, J/kg
$v$ velocity, m/s

Abbreviations

$\dot{V}$ volumetric flow rate, m$^3$/s DB dry bulb
$W$ channel width, m DP dew point
$\dot{W}$ power consumption, W FL flow rate
MVC mechanical vapor compression

Greek symbols

$\delta$ thickness, mm RB relative humidity

1 Introduction

With the advent of digitization and electrification, there is a drastically increasing demand for sensible cooling of electrical and electronic systems, such as data centers and battery packs [1]. Currently, air conditioning systems heavily rely on mechanical vapor compression chillers (MVCs), which can occupy 50% of the building energy demand [2]. The intensive energy consumption of MVC, as well as indirect emissions of greenhouse gases, hinders the sustainable deployment of cooling systems towards a net-zero carbon emission future. In contrast, evaporative cooling technologies demonstrate great potential in green and efficient sensible cooling applications [3, 4], owing to its advantages in high coefficient of performance (COP) and eradication of the needs for compressor and chemical refrigerants [5]. Therefore, evaporative cooling systems have drawn wide attention to developing the next-generation air conditioning systems [6, 7].

For evaporative cooling, the product air temperature for cooling purpose is critical to its performance. Conventional direct/indirect evaporative cooling is limited by the wet bulb (WB) temperature of the inlet air [3]. To break through this bottleneck, dew point evaporative cooling, also named as “the Maisotsenko cycle (M-cycle)”, has been proposed to enable the air cooling towards its dew point (DP) temperature [8, 9].

However, due to the challenges in manufacturing and material development, early efforts to establish a dew point evaporative cooler were not successful and rarely reported. A pioneering work was carried out by Hsu et al. [10] to show that the dew point evaporative coolers could achieve the
maximum WB effectiveness of 1.30. Afterwards, the idea of dew point evaporative cooling did not
draw much attention from the research community until Coolerado Corporation™ launched its first
cross-flow M-cycle cooler [11]. The commercially available cooler was extensively tested by Elberling
[12], Zube and Gillan [13]. Their results demonstrated that the cooler could deliver the product air
temperature at below its inlet WB temperature with an average COP above 9.0. Following the
experimental investigation, Anisimov and Pandelidis et al. [14, 15] came up with a comprehensive ε-
NTU model to study the temperature and humidity distributions in the M-cycle cooler. They extended
the use of M-cycle cooler for heat recovery by taking the room return air as the secondary airflow into
the cooler [16]. With this arrangement, the primary air could be cooled below its dew point when vapor
condensation occurred in the air channels. Rogdakis et al. [17] conducted an in-depth investigation of
the M-cycle cooler under the hot and arid Mediterranean climate. Their experiments facilitated an
adjustable mass flow ratio of supply to working air, and the WB effectiveness and specific water
consumption were ascertained to lie in 0.97–1.15 and 2.5–3.0 kg/kWhc. In addition, Jradi et al. [18]
fabricated a similar cross-flow cooler to the commercial M-cycle cooler. Their cooler could achieve
the 17.3 °C outlet temperature, 1054 W cooling capacity and 14.2 COP, when the supply air
temperature and relative humidity (RH) were at 41.1 °C and 14.5%.

Following the commercialization of cross-flow coolers, many new designs for dew point
evaporative cooling were put forward to improve its performance [20, 21]. Kashyap et al. [22]
compared eight possible cooler configurations, including four parallel/counter-flow and four cross-
flow coolers with different flow directions of primary air, secondary air and water flow. Their
simulations showed that one counter-flow configuration (secondary air and water flow in parallel) and
one cross-flow configuration (primary air and water flow in parallel) obtained the best results of
cooling effectiveness and energy efficiency under most operating conditions. Liu et al. [23] developed
a high-efficiency dew point evaporative cooler. By use of corrugated plates, the cooling effectiveness
of the cooler was 10% higher than a flat-plate cooler. Jia et al. [24] examined two counter-cross-flow
coolers, where the wet channels of one cooler was made of polystyrene board and nylon fiber while
the other was made of aluminum foil. The polystyrene+nylon fiber cooler outperformed the latter with
the DP effectiveness and COP spanning 0.47–0.49 and 10.1–13.8, respectively. Duan et al. [25]
suggested a hybrid dew point evaporative and MVC air conditioning system. Their numerical
simulations indicated that the hybrid system could save 38.2 % electricity consumption and 28.5%
carbon emissions from a stand-alone MVC system. Besides brainstorming different possible cooler
configurations, special attentions were paid to dig into the optimal cooling performance of a well-
established cooler architecture [26]. This is normally accomplished by developing a multi-objective
algorithm [27] that is integrated with thermodynamic models [28] or data-driven models [29, 30].

It has been widely acknowledged that the dew point evaporative cooler delivers promising
performance in accomplishing sub-wet bulb cooling with high energy efficiency. However, nearly all
of the existing work on dew point evaporative cooling merely investigates the performance at steady
states, while its dynamic behaviors under continuous inlet variations are not clear and have not be well
documented. In particular, if the transient response of the cooler is not rapid enough, the overall cooling
performance can deviate from its steady-state rating over a long period. Disturbances in the operating
conditions, such as water spray [31-33], can also destroy the established physical fields in the cooler
and perturb the cooling output. Thus, while enhancing the cooler’s steady-state performance remains
a key interest, there is a large impetus to study its transient response.

Herein we propose an experimental study on the transient response of the dew point evaporative cooler and its critical impact on the time-average cooling performance during dynamic operations. A counter-flow dew point evaporative cooler is specifically tested under zero-state and step responses. It is observed that the response time of the cooler is much longer than that is expected under constant inlet conditions. As a consequence, the time-average cooling performance during transient responses is far below its final steady-state performance. Unfortunately, this phenomenon cannot be predicted by the early established transient model [34], where new physical mechanisms have to be considered to address this challenge. Therefore, a modified lumped parameter cooler model is developed, during which an air mixing process in the dry channel is identified to be responsible for the slow development of temperature and humidity distributions in the cooler. A new partial differential exergy model for the dew point evaporative cooler is constructed to further investigate this finding. In addition, the cooler’s steady-state output is extensively explored under various test conditions and the acquired data are used to validate the modified model. Afterwards, a sensitivity analysis can be carried out to gain insights into the relative importance of the input parameters in determining its transient and steady-state cooling performance, as well as the directions to further optimize their values.

This work elucidates that the transient characteristics of a dew point evaporative cooler can be as important as that at a steady state. Hence, both of the transient and steady-state performance should be judiciously considered in the cooler design. A cooler optimization algorithm can be developed in the future that simultaneously considers the cooler output at steady and unsteady states.

2 Experimental method

2.1 Test apparatus

A dew point evaporative cooler with a counter-flow configuration was designed and shown in Fig. 1(a). The supply air is directed into the dry channels of the cooler and is separated into two portions at the dry channel exits. The major portion is delivered as the product air for cooling, and the rest is employed as the working air and sucked into the wet channels with a reverse direction. The working air in the wet channels can stimulate water evaporation and is finally purged from the channel exits located at one side of the cooler. To fabricate the cooler, as shown in Fig. 1(b), acrylic material was used to form the frame of the air channels, and 200 µm thick PET sheets were employed as the channel plates. In the wet channels, a 200 µm thick natural fiber layer was coated on the channel plate surface to absorb and retain water from a supply water tank at one side of the cooler. The complete cooler consists of five dry and wet channel pairs, and the size of each channel is 0.6 m × 0.15 m × 5.0 mm (L×W×H).

(a) (b)
Fig. 1. The dew point evaporative cooler: (a) cooler design; (b) cooler fabrication; (c) schematics of the test system.

A test system was constructed for the dew point evaporative cooler, as illustrated in Fig. 1(c). The apparatus was placed in a steady indoor environment with about 24.0 °C and 60% RH. The cooler was completely covered by a thick layer of Armaflex material to minimize the heat exchange with the ambient. The supply air temperature and humidity were manipulated by a thermal chamber. An air blower (AFB series, Delta Electronics) was installed at the cooler inlet to regulate the supply air flow rate. The power consumption of the air blower is measured by a portable power meter. The product air flow rate and the consequent working air ratio were adjusted via an air damper at the dry channel outlet.

To obtain the air properties, RTD temperature sensors and air velocity meters were employed to measure the dry bulb (DB) temperatures, WB temperatures and air flow rates of the supply, working and product air streams. During testing, these sensors were connected with a data logger (34970A, Agilent) to collect the experimental data. The sample time of the data logger was set to 2 seconds. The specifications of the sensors used in the experiments are provided in Table 1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Type</th>
<th>Supplier</th>
<th>Range</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature</td>
<td>RTD</td>
<td>Omega Engineering</td>
<td>-29–100 °C</td>
<td>1/10 DIN</td>
</tr>
<tr>
<td>Velocity</td>
<td>Hot wire velocity meter</td>
<td>Omega Engineering</td>
<td>0–10.16 m/s</td>
<td>±2.0% full scale</td>
</tr>
<tr>
<td>Power</td>
<td>Portable power meter</td>
<td>Energetic</td>
<td>0.1–3120 W</td>
<td>±2.0%</td>
</tr>
</tbody>
</table>

Before conducting an experiment, the temperature-humidity climatic chamber was adjusted to preset air conditions and the cooler water tank was filled to ensure a fully wet surface in the wet
channels. The datalogging system and sensor readings were checked prior to start recording. To begin
a test, the datalogging system was firstly turned on, and the air blower and damper were rapidly
regulated to reach an expected supply air flow rate and working air ratio. Each experiment would
continue until a specific objective was met. The test conditions of the dew point evaporative cooler are
listed in Table 2, and the nominal values of the supply air conditions were set at 30.0 °C and 13.3 g/kg
(50% RH).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Nominal value</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Supply air temperature (°C)</td>
<td>30.0</td>
<td>28.0–35.0</td>
</tr>
<tr>
<td>Supply air humidity (g/kg)</td>
<td>13.3</td>
<td>11.0–22.0</td>
</tr>
<tr>
<td>Channel length (m)</td>
<td>0.6</td>
<td>–</td>
</tr>
<tr>
<td>Channel height (mm)</td>
<td>5.0</td>
<td>–</td>
</tr>
<tr>
<td>Supply air velocity (m/s)</td>
<td>1.50</td>
<td>0.90–2.20</td>
</tr>
<tr>
<td>Working air ratio (–)</td>
<td>0.50</td>
<td>0.20–0.70</td>
</tr>
</tbody>
</table>

### 2.2 Performance evaluation

To evaluate the dew point evaporative cooler and predict its performance for large-scale operations,
several quantitative parameters are necessary. The most common parameter is the dew point
effectiveness ($\varepsilon_{dp}$), which measures sensible cooling (temperature reduction) with reference to its
maximum potential at DP temperature, expressed as

$$
\varepsilon_{dp} = \frac{T_s - T_{p}}{T_s - T_{s,dp}}
$$

(1)

where $T_s$ and $T_{s,dp}$ are the supply air temperature and its dew point at the dry channel inlet, $T_p$ is the
product air temperature at the dry channel outlet.

The cooling capacity ($\dot{Q}$) of a cooler is defined as the enthalpy difference between the supply air
and product air

$$
\dot{Q} = \rho_s \dot{V}_p \left( h_s - h_p \right)
$$

(2)

where $\rho_s$, $\dot{V}$ and $h$ are the air density, volumetric flow rate and air enthalpy, respectively.

The energy efficiency of the cooler or COP, is defined as the ratio of the cooling capacity to the
electrical power consumption ($\dot{W}$),

$$
\text{COP} = \frac{\dot{Q}}{\dot{W}}
$$

(3)

### 2.3 Experiment verification

Prior to data analysis, the acquired measurements were checked to examine the accuracy of the
test system. Firstly, the uncertainties of the objective variables \((y)\) can be calculated from the following expression [35]

\[
\frac{\Delta y}{y} = \sqrt{\sum \left( \frac{1}{y \frac{\partial y}{\partial x_i}} \Delta x_i \right)^2}
\]

(4)

where \(x_i\) are the variables measured in the tests, \(\Delta x_i\) are their experimental uncertainties, and \(\Delta y\) is the uncertainty of an objective variable.

The average uncertainties of the objective variables are listed in Table 3.

<table>
<thead>
<tr>
<th>DP effectiveness</th>
<th>Cooling capacity</th>
<th>COP</th>
<th>Exergy</th>
</tr>
</thead>
<tbody>
<tr>
<td>±0.70%</td>
<td>±4.1%</td>
<td>±7.0%</td>
<td>±5.0%</td>
</tr>
</tbody>
</table>

A mass and energy balance exercise was judiciously conducted to ascertain that no physical problems were embedded in the experimental setup, such as air leakage, equipment breakdown, and inaccurate reading. Ideally, the inlet and outlet mass flow rates and energy should be equal, yielding

\[
\dot{m}_s + \dot{m}_f = \dot{m}_p + \dot{m}_w
\]

(5)

\[
\dot{m}_s h_s + \dot{m}_f h_f = \dot{m}_p h_p + \dot{m}_w h_w
\]

(6)

where \(\dot{m}\) denotes the mass flow rate.

In Eqn. (6), as the air enthalpy terms can have a large magnitude, it makes any potential imbalance between the inlet and outlet insignificant and difficult to identify. Thus the equation is rearranged as

\[
\dot{m}_p (h_s - h_p) = \dot{m}_w (h_w - h_s) - \dot{m}_f h_f
\]

(7)

where the left side denotes the energy reduced in the product air while the right side represents the energy absorbed by the working air from the supply air.

Thus the mass and energy balance errors are obtained as follows

\[
Er_m = \frac{\dot{m}_{in} - \dot{m}_{out}}{\dot{m}_{in}} \times 100\%
\]

(8)

\[
Er_e = \frac{\dot{E}_l - \dot{E}_r}{\dot{E}_l} \times 100\%
\]

(9)

where \(\dot{m}_{in} = \dot{m}_s + \dot{m}_f\), \(\dot{m}_{out} = \dot{m}_p + \dot{m}_w\), \(\dot{E}_l = \dot{m}_p (h_s - h_p)\), \(\dot{E}_r = \dot{m}_w (h_w - h_s) - \dot{m}_f h_f\) and \(\dot{E}\) is the energy transfer rate.

Fig. 2 presents the mass and energy balance errors from a typical experiment when the cooler was tested for around two hours. The maximum error for the mass balance is within ±5.0%, while it is within +10.0% to −15.0% for the energy balance. The time-average errors of the mass and energy balances are +1.7% and −2.0%, respectively. Therefore, it is apparent that there are no major inconsistencies in the experimental data and they can be used for performance analysis.
The configuration of the dew point evaporative cooler allows to cool the supply air without humidity addition. The humidity of the product air, however, may increase due to potential water leakage and water vapor permeation in practical cooling systems. When this occurs, the working principle behind indirect evaporative cooling will no longer satisfy. To avoid such a problem, the humidity of each air stream is shown in Fig. 3(a). Obviously, there is a slight difference in air humidity between the dry channel inlet and outlet. The product air humidity is found to be lower than that of the supply air, with a maximum discrepancy within ±2.9%. Thus the assumption of fairly constant air humidity along the dry channel is valid. It is also clear from Fig. 3(b) that water film or droplets can form on the surfaces of the air pipe and sensor probes as the exhausted working air is saturated. This finding is consistent with our earlier findings [36]. When the air channel is longer than a certain value, there is a tendency for the working air to saturate.

Fig. 2. Mass and energy balance errors: (a) mass balance; (b) energy balance.

Fig. 3. Humidity verification of the evaporative cooler: (a) air humidity profiles of the three air streams; (b) water condensation at the outlet of the wet channels.
3 Mathematical model

3.1 First law of thermodynamics

In most existing studies, the performance of dew point evaporative coolers were examined under steady-state conditions where constant inlet air conditions were assumed. However, in real applications, the inlet air temperature and humidity, often than not, varies throughout the operating period, as can be seen from Fig. 3(a). The air with different states will undergo a mixing process in the dry air channels. If the design parameters are not well defined, the air mixing phenomenon is a slow process, as demonstrated in the transient-state experiments (see Section 4). Therefore, the energy and mass balance equations for the dry channel ought to be updated to consider the mixing process. These circumstances deviate the cooler from its ideal steady states, where a steady-state mathematical model would fail and instead a transient model is essential. Accordingly, a modified transient lumped parameter model is developed in this work, based on differential control volumes with infinitesimal length along the flow direction for the respective supply air, working air and water film, as shown in Fig. 4. Several assumptions are introduced to simplify the model:

1. Owing to the small channel height and large channel aspect ratio \((W/H)\), air distributions along the channel width and height are assumed uniform so that the model geometry can be reduced from 3D to 1D to improve the computational efficiency.

2. A thin layer of static water film covers the entire wet channel surface.

3. The thermodynamic state of the air is assumed to be homogenous in each small control volume.

4. The pressure drops of the air streams are neglected as their momentum balance are not considered.

5. Heat capacity and thermal resistance of the channel plate are neglected.

6. Heat exchange between the cooler and the surroundings is negligible.

![Image](image_url)

**Fig. 4.** Energy and mass balances of a small control volume for: (a) supply air; (b) working air; (c) water film.

The governing equations for each control volume are derived from the principles of energy and mass conservation. In the dry channel, as the air mixing process exists, the air humidity is no longer a constant. The internal energy and enthalpy of the air are functions of both temperature and humidity, while a mass balance equation is incorporated to calculate the water vapor species of each volume. Then the equations for the dry channel are expressed as
\[ \rho_d c_{v,d} \frac{\partial T_d}{\partial t} + u_v(T_d) \frac{\partial \rho_{v,d}}{\partial t} = k_d \frac{\partial^2 T_d}{\partial x^2} - \frac{2 \bar{h}_d}{H} (T_d - T_f) - \rho_d c_{p,d} \nu_d \frac{\partial T_d}{\partial x} - \nu_d \frac{\partial \rho_{v,d}}{\partial x} \] (10)

\[ \frac{\partial \rho_{v,d}}{\partial t} = D_{va} \frac{\partial^2 \rho_{v,d}}{\partial x^2} - \nu_d \frac{\partial \rho_{v,d}}{\partial x} \] (11)

where \( T \) and \( \rho_v \) are the temperature and vapor density (humidity) to be solved, \( \rho_d, c_v, c_p, k, \bar{h}, D_{va} \) and \( \nu \) are air density, specific heat at constant volume, specific heat at constant pressure, thermal conductivity, convective heat transfer coefficient, water vapor diffusivity in air and air velocity, respectively. The internal energy \( u(T) \) and enthalpy \( h(T) \) are functions of temperature.

For the working air, the coupled heat and mass transfer phenomenon during water evaporation is captured via the energy and mass balance equations as follows [34]

\[ \rho_a c_{v,w} \frac{\partial T_w}{\partial t} + u_v(T_w) \frac{\partial \rho_{v,w}}{\partial t} = k_w \frac{\partial^2 T_w}{\partial x^2} - \frac{2 \bar{h}_w}{H} (T_w - T_f) + \rho_a c_{p,w} \nu_w \frac{\partial T_w}{\partial x} + \nu_w h_v(T_w) \frac{\partial \rho_{v,w}}{\partial x} + \frac{2 \bar{h}_w}{H} h_v(T_f)(\rho_{f,sa} - \rho_{v,w}) \] (12)

\[ \frac{\partial \rho_{v,w}}{\partial t} = D_{va} \frac{\partial^2 \rho_{v,w}}{\partial x^2} + \nu_w \frac{\partial \rho_{v,w}}{\partial x} + \frac{2 \bar{h}_m}{H} (\rho_{f,sa} - \rho_{v,w}) \] (13)

where \( \nu_w = \rho_{v,d} \), \( \bar{h}_m \) and \( \rho_{f,sa} \) represent the convective mass transfer coefficient and vapor saturation pressure on the water surface, respectively.

The governing equations for the water film incorporate heat conduction and heat and mass convection with the air streams, expressed as

\[ \rho_f c_{p,f} \frac{\partial T_f}{\partial t} = k_f \frac{\partial^2 T_f}{\partial x^2} + \frac{\bar{h}_d}{\delta} (T_d - T_f) + \frac{\bar{h}_w}{\delta} (T_w - T_f) + \frac{\bar{h}_m}{\delta} (\rho_{f,sa} - \rho_{v,w})(c_{p,f} T_f - h_v(T_f)) \] (14)

where \( \delta \) is the water film thickness.

The boundary conditions of the model equations are listed in Table 4. Furthermore, if the time-dependent terms in the equations are neglected, the transient model can be converted to a steady-state format [36]. In addition, the correlations for the convective heat and mass transfer coefficients in Eqn. (10)–(14) and the psychrometric and transport properties of moist air, saturated water and vapor can be found in some previous work [34, 37, 40-42].

### Table 4 Initial and boundary conditions for the simulation.

<table>
<thead>
<tr>
<th>Supply air</th>
<th>Working air</th>
<th>Water film</th>
</tr>
</thead>
<tbody>
<tr>
<td>( t \leq 0, 0 \leq x \leq L : \quad T_d = T_0 )</td>
<td>( t \leq 0, 0 \leq x \leq L : \quad T_w = T_0, \rho_v = \rho_{v,sa}(T_0) )</td>
<td>( t \leq 0, 0 \leq x \leq L : \quad T_f = T_0 )</td>
</tr>
<tr>
<td>( t &gt; 0, x = 0 : \quad T_d = T_y, \quad \frac{\partial T_d}{\partial x} = 0 )</td>
<td>( t &gt; 0, x = L : \quad T_w = T_d, \rho_v = \rho_{v,d} )</td>
<td>( t &gt; 0, x = 0 : \quad \frac{\partial T_f}{\partial x} = 0 )</td>
</tr>
<tr>
<td>( \frac{\partial T_w}{\partial x} = 0, \frac{\partial \rho_{v,w}}{\partial x} = 0 )</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

11
The air humidity ratio ($\omega$) is used to calculate the water vapor density in the above equations. It is computed using the measured DB ($T_{db}$) and WB ($T_{wb}$) temperatures as below [37]

$$\omega = \frac{(2501 - 2.326 T_{wb}) \omega_{sa} - 1.006 (T_{db} - T_{wb})}{2501 + 1.86 T_{db} - 4.186 T_{wb}} \tag{15}$$

where $\omega_{sa}$ is the saturation humidity at the WB temperature.

The saturation pressure of the water vapor ($P_{v,sa}$) on the water film surface (0–200 °C) is given by [37]

$$\ln(P_{v,sa}) = \frac{C_1}{T_f} + C_2 + C_3 T_f + C_4 T_f^2 + C_5 T_f^3 + C_6 \ln T_f \tag{16}$$

where $T_f$ is the water film temperature in K, and $C_1 = -5.800 \times 10^3$, $C_2 = 1.3914993$, $C_3 = -4.8640239 \times 10^{-2}$, $C_4 = 4.1764768 \times 10^{-5}$, $C_5 = -1.4452093 \times 10^{-8}$, $C_6 = 6.5459673$.

The relationship between humidity ratio and water vapor pressure ($P_v$) is described as [38]

$$\omega = 0.621945 \frac{P_v}{P - P_v} \tag{17}$$

where $P$ is the atmospheric pressure.

### 3.2 Second law of thermodynamics

Based on the cooler model from first law of thermodynamics, a partial differential exergy model can be established to study the irreversibility occurring in the dew point evaporative cooler, in particular during air mixing. As shown in Fig. 5, an exergy balance can be established on the same control volume of the supply air, working air and water film, respectively.

![Fig. 5. Exergy balance of a small control volume for: (a) supply air; (b) working air; (c) water film.](image)

For each control volume, the exergy change is equal to the net exergy transfer into the volume
associated with the heat and mass transfer from/into other volumes subtracting the exergy destroyed within the volume. Therefore, the exergy balance equations for the three domains can be formulated as follows

\[
\frac{\partial e_{x,d}}{\partial t} = \left(1 - \frac{T_0}{T_d}\right) \left(\frac{k_d}{\rho_a H} \frac{\partial^2 T_d}{\partial x^2} - \frac{2\tilde{h}_d}{\rho_a H} \left(1 - \frac{T_0}{T_f}\right) (T_d - T_f) - v_d \frac{\partial e_{x,d}}{\partial x} + \dot{e}_{x,d,de}\right)
\]

(18)

\[
\frac{\partial e_{x,w}}{\partial t} = \left(1 - \frac{T_0}{T_w}\right) \left(\frac{k_w}{\rho_a H} \frac{\partial^2 T_w}{\partial x^2} - \frac{2\tilde{h}_w}{\rho_a H} \left(1 - \frac{T_0}{T_f}\right) (T_w - T_f) + v_w \frac{\partial e_{x,w}}{\partial x} + \frac{2\tilde{h}_m}{\rho_f \delta} \left(\rho_{f,sa} - \rho_{v,w}\right) e_v - \dot{e}_{x,w,de}\right)
\]

(19)

\[
\frac{\partial e_{x,f}}{\partial t} = \left(1 - \frac{T_0}{T_f}\right) \left(\frac{k_f}{\rho_f \delta} \frac{\partial^2 T_f}{\partial x^2} + \frac{h_f}{\rho_f \delta} \left(1 - \frac{T_0}{T_a}\right) (T_a - T_f) + \frac{h_w}{\rho_f \delta} \left(1 - \frac{T_0}{T_w}\right) (T_w - T_f) - \dot{e}_{x,f,de}\right)
\]

(20)

where \( e_x \) and \( e_{x,de} \) denote specific exergy and specific exergy destruction, respectively. \( T_0 \) is the reference temperature at a dead state.

In Eqn. (20), the amount of water that evaporates and leaves the water film will immediately be compensated by the same amount of water absorbed from the water tank via the wick material. Hence the net exergy transfer due to the water mass flow is zero.

The specific flow exergy of moist air, saturated water vapor and saturated liquid water can be calculated from the temperature and humidity fields using the following equations [39]

\[
e_{x,a} = (c_{p,a} + \omega c_{p,v}) T_0 \left(\frac{T}{T_0} - 1 - \ln \frac{T}{T_0}\right) + R_a T_0 \left[\left(1 + \dot{\omega}\right) \ln \frac{1 + \dot{\omega}}{1 + \dot{\omega}_0} + \dot{\omega} \ln \frac{1 + \dot{\omega}_0}{1 + \dot{\omega}}\right]
\]

(21)

\[
e_{x,v} = h_v(T) - h_v(T_0) - T_0 s_v(T) + T_0 s_v(T_0) - R_v T_0 \ln \phi_0
\]

(22)

\[
e_{x,f} = h_f(T) - h_v(T_0) - T_0 s_f(T) + T_0 s_f(T_0) - R_f T_0 \ln \phi_0
\]

(23)

where \( \dot{\omega} = 1.608 \omega, \ R_a = 0.287 \text{ kJ/(kg·K)} \) and \( R_v = 0.461 \text{ kJ/(kg·K)} \). \( T_0, \ \omega_0, \ \phi_0 \) are the temperature, humidity and relative humidity at a predefined dead state. \( s(T) \) is the entropy as a function of temperature.

It can be seen that the specific flow exergy of moist air is a function of temperature and humidity, while the saturated liquid water’s specific flow exergy is only a function of temperature. Hence, the following equations can be obtained

\[
\frac{\partial e_{x,m}}{\partial t} = \frac{\partial e_{x,m}}{\partial T} \frac{\partial T}{\partial t} + \frac{\partial e_{x,m}}{\partial \omega} \frac{\partial \omega}{\partial t}
\]

(24)

\[
\frac{\partial e_{x,m}}{\partial x} = \frac{\partial e_{x,m}}{\partial T} \frac{\partial T}{\partial x} + \frac{\partial e_{x,m}}{\partial \omega} \frac{\partial \omega}{\partial x}
\]

(25)

\[
\frac{\partial e_{x,f}}{\partial t} = \frac{\partial e_{x,f}}{\partial T} \frac{\partial T}{\partial t}
\]

(26)
Rearranging Eqn. (21)–(26) yields

$$\frac{\partial e_{x,a}}{\partial T} = \left( c_{p,a} + \omega c_{p,v} \right) \left( 1 - \frac{T_0}{T} \right), \quad \frac{\partial e_{x,a}}{\partial \omega} = c_{p,v} T_0 \left( \frac{T}{T_0} - 1 - \ln \frac{T}{T_0} \right) + 1.608 R_a T_0 \left( \ln \frac{1 + \dot{\omega}_0}{1 + \dot{\omega}} + \ln \frac{\dot{\omega}}{\dot{\omega}_0} \right)$$

(27)

$$\frac{\partial e_{x,f}}{\partial T} = \frac{\partial h_f}{\partial T} - T_0 \frac{\partial s_f}{\partial T}$$

(28)

where \(\frac{\partial h_f}{\partial T} = c_{p,f} \cdot \)

In Eqn. (18)–(20), the specific exergy destruction (\(e_{x,de}, e_{x,ve}, e_{x,de} \)) is the unknown variable to be solved, of which the magnitude quantifies the irreversibility in the cooling process. On the other hand, the dew point evaporative cooler also satisfies a general exergy balance equation as

$$\dot{E}_X_s + \dot{E}_X_f - (\dot{E}_X_p + \dot{E}_X_w) = \dot{E}_X_{de}$$

(29)

To solve the partial differential equations, each model domain is separated into 20 spatial cell elements, leading to a grid size of 0.03 m along the x direction. A mesh independence analysis is performed in Fig. 6 to show that the simulated temperature results only vary by 0.1–0.2 °C if the quantity of cell elements is increased from 20 to 100. Hence, the number of cell elements is determined to be 20. In addition, the time step of the transient simulation is set to a small enough value at 0.01 s, which balanced the computational efficiency and simulation stability. The governing equations are simultaneously discretized with a finite difference method and solved in a MATLAB environment. The governing equations from the first law of thermodynamics are first solved for the temperature and humidity fields. Afterwards, the exergy balance equations can be solved for the exergy destruction at different temporal steps and spatial locations.
Fig. 6. Mesh independence analysis. The model simulation is carried out with nominal test conditions in Table 2 at steady state.

3.3 Sensitivity analysis

After a general mathematical model is established, a sensitivity analysis can be conducted to gain important insights into the dominance of major input variables in its cooling performance (output variables). Apart from the DP effectiveness, cooling capacity and COP which represent the typical steady-state performance, a quantitative parameter is required to evaluate the cooler’s transient performance. It is later demonstrated in Section 4 that the dew point evaporative cooler is a first-order system where the product air temperature under a zero-state response tends to be an exponential decay function, expressed as

\[ T_p = T_{ss} + ae^{-\frac{t}{t_c}} \]  

where \( T_{ss} \) is the temperature at steady state and \( t_c \) is time constant.

The time constant denotes the time required for the product air temperature to reach 63.2% of its final steady state from its initial state and can be employed to evaluate the speed of the transient response.

Therefore, the aforementioned four parameters are defined as the objective variables in the sensitivity analysis. The input variables that can be independently adjusted in the cooler, are supply air velocity, working air ratio, channel height and channel length. The input and objective variables are summarized as

\[ x_i \in \{v_s, r, H, L\}, \quad y_i \in \{t_c, \varepsilon_{ap}, \dot{Q}, \text{COP}\} \]  

The sensitivity of an objective variable can be calculated by its partial derivative with respect to each input variable at its current state. The first-order partial derivative of each objective variable eventually forms a Jacobian matrix, with its element defined as follows [43]

\[ J_{ij} = \frac{\partial y_i}{\partial x_j} \]

As there are no explicit functions between the objective and input variables, the partial derivatives are numerically computed via the cooler model, when each input variable is perturbed by a small amount, e.g., +5.0%. Ideally, the dominant factor is expected to have the largest value of partial derivative and can be thereby determined. However, it is obvious that the input variables in Eqn. (31) come with different orders of magnitude, some of which are considerably less than the others, such as \( v_s \) and \( r \). Following this, the partial derivatives with respect to a small input variable may be greater than others, even if it has a minor effect on the objective parameters. Such an outcome can potentially mislead the sensitivity analysis and is therefore not desired. To avoid this problem, a relative sensitivity is adopted instead, defined as [44, 45]

\[ J'_{ij} = \frac{\partial \ln y_i}{\partial \ln x_j} = \frac{x_j}{y_i} \frac{\partial y_i}{\partial x_j} \]
4 Transient-state performance

4.1 Zero-state response

The transient response of the cooler was extensively studied. Fig. 7 shows an example of the cooler’s zero-state response, where the cooler was in thermal equilibrium with the ambient before testing. The air temperature and humidity was about 24.4 °C and 18.2 g/kg everywhere in the cooler. The temperature-humidity chamber was set at 30.0 °C and 14.5 g/kg (55% RH). To start the experiment, the supply air flow was rapidly increased from 0 m/s to 1.50 m/s, and the working air ratio was kept at 0.50 during testing. The rated fan power was about 3.6 W.

Fig. 7(a) shows the cooler’s transient temperature responses. The experimental data were recorded at two-second intervals. It was observed that inlet air conditions to the cooler did not immediately reach the set point, although the supply air conditions were constant in the chamber. Instead, there was a gradual air mixing process in the piping between the chamber and cooler. Consequently, the supply air gradually approached the preset conditions. Furthermore, the development of product air temperature was also sluggish. The product air temperature was above the inlet WB temperature in the first 800 s, and after t=1200 s, it became stable with the variation less than 0.2 °C.

The above phenomena indicate that the transient response of the cooler is a rather slow process. In a previous work [34], it was found that the settling time of a cooler under different supply air conditions is within 300 s. However, this was based on an ideal situation of constant supply air conditions. Due to variations in the inlet air, the cooler requires an extended response time to settle. Therefore, understanding the transient behavior is important as far as a real application of dew point evaporative cooling is concerned. The correlations for the supply and product air temperatures can be derived based on the experimental data, yielding the following exponential decay functions

\[ T_s = 30.03 - 5.46e^{-t/900.90} \ (°C) \]  \hspace{1cm} (34)

\[ T_p = 21.63 + 2.80e^{-t/596.53} \ (°C) \]  \hspace{1cm} (35)

The perfect fitting of the air temperatures suggests the dew point evaporative cooler follow the characteristic of a first-order system [46]. The time constants in this test were approximately 901 s and 397 s for the supply and product air temperatures, respectively.

The associated cooling performance is further elaborated in Fig. 7(b)–(d). Apparently, the DP effectiveness, cooling capacity and COP increased gradually from the zero point to 0.80, 5.4 W/channel and 7.5, respectively. Again, due to the air mixing process, the cooler output could not immediately approach its steady state, especially for the cooling capacity and COP. The time-average DP effectiveness, cooling capacity and COP over 2500 s are 0.69, 4.0 W/channel and 5.5, which are 13.8%, 26.4% and 26.4% less than its final steady-state performance.

In addition, the experimental supply air conditions were used to simulate the cooling performance and the results are presented in Fig. 7. It is evident that the modified transient model can accurately capture the transient developments of product air temperature, DP effectiveness, cooling capacity and COP. The maximum discrepancies for the four parameters are within ±2.0%, ±5.0%, ±6.0% and ±7.0%, respectively.
Fig. 7. Zero-state response of the dew point evaporative cooler: (a) temperature; (2) DP effectiveness; (c) cooling capacity/channel; (d) COP.

The impact of the air mixing process could be verified by examining the irreversibility in the cooler, so an exergy analysis of the transient cooling process in Fig. 7 was carried out. Fig. 8(a) shows the dynamic exergy balance in the cooler following Eqn. (29). The preset supply air conditions, namely, 30.0 °C and 14.5 g/kg, were chosen as the dead state for exergy calculation. It was observed that the supply water accounted for the largest amount of exergy into the cooler, where its exergy was transferred to the product air via water evaporation and heat transfer during the cooling process. The supply air possessed a small amount of exergy at the beginning of the test, but its exergy gradually reduced to zero as it approached the dead state. The working air contained a constant exergy content which was eventually lost to the environment at the wet channel exit. Hence, the exergy destruction, as the difference between total exergy input and output, could vary from 70.0% at the beginning to 29.4% when the cooling process started to stabilize. This indicates that a large degree of irreversibility could occur when the cooler is at an unsteady state.

A further breakdown of the exergy destruction from different domains can be seen in Fig. 8(b). Apparently, the major exergy destruction came from the wet channel due to the coupled heat and mass
transfer. On the contrary, the local exergy destruction in the water film was negative as it transferred more exergy to the working air than that it absorbed from the supply air. More importantly, the exergy destruction contributed by the dry channel in Fig. 8(b) is marginal, which seems to speculate a small irreversibility. To confirm this observation, the specific exergy destruction of the air and water at different spatial locations along the channels is plotted in Fig. 8(c). The time step at 2500 s was selected, while the results at other time steps were found to be similar. It is clear that the specific exergy destruction of the supply air at the dry channel entrance was significantly large, as attributed to the air mixing process where the inlet air with a changing thermodynamic state exchanging heat and mass with the substances in the dry channels. The specific exergy destruction was lowered from ca. 400 W/kg to negative values after half channel length (x>0.3 m), and eventually reached ca. -63 W/kg at the end of dry channel. This infers that the air mixing process was completed, and the negative specific exergy destruction was dominated by the evaporative cooling that increased the air exergy along the dry channel. Consequently, the large variation of the exergy destruction from positive to negative values at different dry channel locations counterbalanced and yielded a small net exergy destruction in Fig. 8(b). Besides, the negative exergy destruction in the dry channel comes at a cost of large exergy destruction in the wet channel to drive the unspontaneous cooling process. It is noticed that the specific exergy destruction of the working air is a convex function with respect to the channel location. The specific exergy destruction was below zero between x=0.30 m and x=0.45 m, while its value was in the order of 100 W/kg at other locations. To explain this finding, temperature distributions of dry channel, wet channel and water film are plotted in Fig. 8(d) for reference. It can be found that the negative specific exergy destruction region lies in where the working air was about to saturate and its temperature was reduced to below the water film temperature. Hence, the working air and water film were close to equilibrium and the heat and mass transfer between them was minute, which induced negligible irreversibility.

(a)  (b)  
(c)  (d)
Fig. 8. Exergy analysis of the zero-state response: (a) general exergy balance; (b) net exergy destruction in different domains; (c) distributions of specific exergy destruction along the channels at 2500 s; (d) temperature distributions along the channels at 2500 s.

4.2 Step response

Apart from the zero-state response, the transient performance of the cooler under a step response was also explored. Fig. 9 presents the results of one sample test. The experiment was performed by introducing a step change in the supply air during a steady-state operation. Initially the inlet air was kept constant at 30.0 °C and 13.5 g/kg, and the product air temperature was about 21.7 °C. The supply air velocity was set at 1.50 m/s with a working air ratio of 0.50. To impose a step change, the inlet air conditions were rapidly adjusted to 36 °C and 15.0 g/kg.

Although the inlet temperature profiles underwent significant variations, the product air temperature remained relatively constant. After \( t = 1600 \) s, temperature variation of the product air was observed to be less than 0.2 °C. However, the DP effectiveness changed dramatically which was principally attributed to the variation of supply air DP temperature. Two peak values (0.81 and 0.94) are observed at \( t = 350 \) s and \( t = 900 \) s. Furthermore, the cooling capacity and COP of the cooler increased rapidly at the beginning phase, while their slopes reduced after 800 seconds. The maximum cooling capacity for a single dry channel was found to be 8.0 W operating with a COP of 11.1. In agreement with the zero-state response, the delay in the product air temperature results in pronounced deviations of the time-average cooling performance from its final state, as seen in Fig. 9(b)–(d).

Similarly, simulation results of the step response are compared with the test results in Fig. 9. The transient lumped parameter model achieves good agreement with the experiment, and the maximum discrepancies of the product air temperature, DP effectiveness, cooling capacity and COP are within ±2.0%, ±4.0%, ±6.0% and ±6.0%. It can also be inferred from the above results that the dynamic behavior of the cooler appears to be irregular and highly depends on the supply air conditions. Therefore, the dynamic performance of the evaporative cooler can only be accurately predicted and analyzed via the transient model with the known inlet air conditions.
Fig. 9. Step response of the dew point evaporative cooler after reaching a steady state: (a) temperature; (b) DP effectiveness; (c) cooling capacity/channel; (d) COP.

Based on the simulated temperature and humidity fields, a similar exergy analysis of the step response can be conducted. As can be seen in Fig. 10, the exergy destruction in the cooler constituted 17.2% to 36.1% of the total exergy input during the step response. Unlike the zero-state response, the exergy destruction in the dry channel was remarkable before $t=1000$ s (Fig. 10(b)) when the cooler was adjusted from its previous steady state to a new state, owing to the abrupt jump in the supply air. The spatial distributions of the specific exergy destruction and temperature are plotted in Fig. 10(c) and (d) at $t=3000$ s. Compared to the zero-state response, the specific exergy destruction in both dry and wet channels was one order of magnitude larger, i.e., in 1000 W/kg. These results reveal that the irreversibility caused by the air mixing process in the dry channel is more pronounced during continuous operation of the cooler, especially when the supply air conditions are unsteady.
Fig. 10. Exergy analysis of the step response: (a) general exergy balance; (b) net exergy destruction in different domains; (c) distributions of specific exergy destruction along the channels at 3000 s; (d) temperature distributions along the channels at 3000 s.

As seen in the zero-state and step responses of the cooler, the air mixing process in the dry channel is crucial to the response time and consequent overall performance of the cooler. An in-depth exergy analysis justifies that the air mixing process mostly happens at the first half of the dry channel (x<0.30 m) and induces a large degree of irreversibility. More importantly, enhancing the cooler’s dynamic performance requires an improvement in its transient time constant. This may in turn increase the irreversibility of the cooling process, as this is against a slow and quasi-equilibrium process which is thermodynamically favored. Hence, the transient response should be judiciously considered in the cooler design and control.

5 Steady-state performance

5.1 Test results under varying conditions

After investigating the transient responses, the steady-state output of the cooler is explored. In this study, a broad range of ambient conditions was created to investigate the capability of the cooler. The
supply air temperature was regulated from 28.0 °C to 35.0 °C, while the air humidity was varied from 11.0 g/kg to 22.0 g/kg. The supply air velocity and working air ratio during testing were controlled their nominal conditions (1.50 m/s and 0.50).

Fig. 11 illustrates the cooler performance under various supply air conditions. ASHRAE Standard 55 [47] recommends a temperature range of 23.5–28.0 °C for thermal comfort in summer. Accordingly, the product air temperatures from the cooler were below 25.0 °C under most conditions, except when the air humidity was above 19.0 g/kg (inlet DP temperature was higher than 25.0 °C). Therefore, the cooler could satisfy the thermal comfort requirement. The corresponding DP effectiveness ranged from 0.67 to 0.94, while averaging at 0.78. The cooling capacity for each dry channel was mostly between 5.0 W and 6.0 W, but it could exceed 7.0 W when the supply air was above 34.0 °C. The COP of the cooler varied from 4.2 to 11.1, depending on the temperature drop and cooling capacity.

![Images of graphs showing air temperature, effectiveness, cooling capacity, and COP](image)

**Fig. 11.** Test results under various supply air conditions: (a) product air temperature; (b) DP effectiveness; (c) cooling capacity/channel; (d) COP. The black and grey domains are bounded without available experimental data.

The influence of the operating conditions, i.e., supply air velocity and working air ratio, on the cooler performance was subsequently studied and summarized in Table 5. The inlet air were controlled
at 30.0–31.0 °C and 16.0–18.0 g/kg. When the supply air velocity was increased from 0.98 m/s to 2.14 m/s, the product air temperature was found to rise from 23.6 °C to 25.0 °C, deteriorating the DP effectiveness. In contrast, the cooling capacity and COP generally increased with the supply air velocity, as larger volume of product air was delivered.

Similarly, when the working air ratio was increased, the DP effectiveness became more favorable, while a maximum cooling capacity and COP were observed at 0.30 working air ratio. This agrees well with several existing studies [20, 48] where an optimal cooling capacity exists at 0.30–0.40 working air ratio. When the ratio continued to rise, more supply air was employed as working air and less product air was delivered, leading to a reduction in cooling capacity and COP.

**Table 5** Test results under different operating conditions.

<table>
<thead>
<tr>
<th>$\nu_s$ (m/s)</th>
<th>$r$ (–)</th>
<th>$T_s$ (°C)</th>
<th>$\omega_s$ (g/kg)</th>
<th>$T_p$ (°C)</th>
<th>$\varepsilon_{dp}$ (–)</th>
<th>$\hat{Q}$ (W/channel)</th>
<th>COP (–)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.98</td>
<td>0.50</td>
<td>30.2</td>
<td>17.0</td>
<td>23.6</td>
<td>0.84</td>
<td>2.8</td>
<td>4.4</td>
</tr>
<tr>
<td>1.20</td>
<td>0.50</td>
<td>30.4</td>
<td>17.0</td>
<td>23.9</td>
<td>0.80</td>
<td>3.4</td>
<td>4.9</td>
</tr>
<tr>
<td>1.50</td>
<td>0.49</td>
<td>30.5</td>
<td>16.8</td>
<td>24.3</td>
<td>0.74</td>
<td>4.2</td>
<td>5.8</td>
</tr>
<tr>
<td>1.86</td>
<td>0.51</td>
<td>30.8</td>
<td>16.6</td>
<td>24.5</td>
<td>0.71</td>
<td>5.0</td>
<td>7.0</td>
</tr>
<tr>
<td>2.14</td>
<td>0.52</td>
<td>30.3</td>
<td>17.2</td>
<td>25.0</td>
<td>0.68</td>
<td>4.7</td>
<td>6.4</td>
</tr>
<tr>
<td>1.49</td>
<td>0.21</td>
<td>30.7</td>
<td>16.1</td>
<td>25.5</td>
<td>0.56</td>
<td>5.4</td>
<td>7.5</td>
</tr>
<tr>
<td>1.47</td>
<td>0.30</td>
<td>30.8</td>
<td>16.4</td>
<td>24.6</td>
<td>0.69</td>
<td>5.6</td>
<td>7.8</td>
</tr>
<tr>
<td>1.49</td>
<td>0.42</td>
<td>30.8</td>
<td>16.1</td>
<td>24.0</td>
<td>0.73</td>
<td>5.2</td>
<td>7.2</td>
</tr>
<tr>
<td>1.46</td>
<td>0.49</td>
<td>30.7</td>
<td>16.5</td>
<td>24.0</td>
<td>0.76</td>
<td>4.4</td>
<td>6.0</td>
</tr>
<tr>
<td>1.54</td>
<td>0.62</td>
<td>30.6</td>
<td>16.6</td>
<td>24.0</td>
<td>0.77</td>
<td>3.5</td>
<td>4.8</td>
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<tr>
<td>1.50</td>
<td>0.69</td>
<td>30.5</td>
<td>17.0</td>
<td>24.1</td>
<td>0.79</td>
<td>2.6</td>
<td>3.6</td>
</tr>
</tbody>
</table>

It should be noted that the dew point evaporative cooler owns superior energy efficiency to a typical conventional air conditioner in most cases [49, 50]. Nonetheless, the cooler COP is still limited at current state due to the inefficient fan power at a small air flow rate. In addition, as shown in Section 4, the steady-state cooler performance determines the final state of the cooler in a transient response, which consequently affects the dynamic output of the cooler. Hence, improving the cooler design will enable a simultaneous enhancement in both of its transient and steady-state thermodynamic performance.

### 5.2 Model validation

Steady-state simulations were conducted to validate the lumped parameter model, where the entire range of test results was employed. Fig. 12 compares the simulated cooler performance with the test results from Section 5.1. Again, the model is capable of capturing the product air temperature under various supply air conditions with a minor discrepancy of ±0.7 °C (Fig. 12(a)) and achieve even better accuracy under changing operating conditions in Fig. 12(b)–(c).

Therefore, the lumped parameter model has demonstrated its competent credibility in predicting the output performance of the dew point evaporative cooler and can be applied to the sensitivity analysis in the ensuing section.
6 Sensitivity analysis

Since the transient and steady-state characteristics of the dew point evaporative cooler have been experimentally and numerically evaluated, the sensitivity of the relevant objective variables can now be analyzed. As stated in Section 3.3, the Jacobian matrix of the objective variables’ partial derivatives is calculated by perturbing each input variable from its nominal value provided in Table 2. The changes in objective variables are then obtained from the cooler model. The results of the absolute ($J_y$) and relative ($J'_y$) sensitivity are shown in Table 6.

<table>
<thead>
<tr>
<th>Table 6 Sensitivity analysis of the dew point evaporative cooler.</th>
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<tbody>
<tr>
<td>Objective variable</td>
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<tr>
<td>----------------------</td>
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<tr>
<td>$t_c$</td>
</tr>
<tr>
<td>$\varepsilon_{dp}$</td>
</tr>
<tr>
<td>$\dot{Q}$</td>
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<tr>
<td>COP</td>
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</table>
The sensitivity of the objective variables can provide a future perspective to improve the cooler design, taking its current design as a baseline. The sign (+/-) of each partial derivative suggests if the objective variable is increasing or decreasing with an input variable, so a decision can be made on varying the input variables to maximize or minimize the objective variable. From the absolute sensitivity, it is observed that the channel height tends to have a large impact on every objective variable, especially the time constant and COP. This is attributed to its small length scale that magnifies its contribution. Instead, the relative sensitivity provides a clear judge of the relevance of the input variables. The normalized partial derivatives with larger magnitude are deemed to play a more important role in the objective variables. Hence, the respective dominant factors of the time constant, DP effectiveness, cooling capacity and COP, which can be manipulated to enhance the cooling performance, are channel length, channel height, working air ratio and channel height. The magnitude of their relative sensitivity at nominal conditions are 0.77, -0.90, -0.88 and 2.03, respectively.

7 Conclusions

A dew point evaporative cooler was designed, engineered and investigated. The transient responses of the cooler was judiciously examined, followed by investigations of its steady-state performance.

It was observed that the supply and product air temperature profiles of the cooler follow exponential decay functions under a zero-state response, which adhered to the characteristics of a first-order thermodynamic system. However, the transient response of the cooler was dramatically slower than one that is subjected to a constant inlet air condition, where large time constants of 901 s and 397 s are found for the supply air and product air temperatures, respectively. Consequently, the time-average cooling performance of the dew point evaporative cooler was 13.8%–26.4% below its final state over the test duration of 2500 s. Similarly, the dew point evaporative cooling process was also not fast enough to exhibit instant changes at the product air outlet under step responses. To simulate these cooling processes, a modified transient lumped parameter mathematical model was developed to identify that an air mixing process in the dry channel was critical to cause the slow transient response. By taking into account this mechanism, the model could simulate the product air temperature, DP effectiveness, cooling capacity and COP within ±2.0%, ±5.0%, ±6.0% and ±7.0% discrepancies, respectively.

To further examine the role of the air mixing process, the irreversibility of the cooling process is quantified by conducting an in-depth exergy analysis. A novel partial differential exergy model revealed that the exergy destruction varied from 70.0% to 29.4% throughout the zero-state response. Although the net exergy destruction from the supply air was marginal, it was found that the specific exergy destruction at the first half dry channel (x<0.3 m) was above 400 W/kg, comparable to that in the working air. This demonstrated that the air mixing process was significant at the dry channel entrance but was completed by the first half of the channel, after which the exergy destruction of the supply air turned negative upon cooling. The specific exergy destruction of the working air was in the order of 100 W/kg at the wet channel entrance and exit, but could reach zero between x=0.30 m and x=0.45 m. This is mainly due to the dynamic thermal equilibrium established between the working air and water film in this region.
Subsequently, the steady-state performance of the cooler was investigated under a variety of test conditions. The product air temperature was in the range of 20.6–26.2 °C, and the consequent DP effectiveness, cooling capacity and COP spanned 0.67–0.94, 3.0–8.0 W/channel and 4.2–11.1, respectively. The steady-state rating projects the ultimate output after the cooling process stabilizes under dynamic operations and thereby affects the time-average cooling performance.

Therefore, both the transient and steady-state characteristics of the dew point evaporative cooler are important and ought to be carefully designed and optimized, owing to the air mixing process under dynamic operations. To gain insights into the relative importance of the input variables in affecting the cooling performance, as well as the directions to manipulate them, a sensitivity analysis was suggested to enable simultaneous enhancements of the cooler’s transient and steady-state performance. The four objective variables (time constant, DP effectiveness, cooling capacity and COP) are observed to be mostly dominated by the channel length, channel height, working air ratio and channel height, with their relative sensitivity at the nominal design conditions to be 0.77, -0.90, -0.88 and 2.03, respectively.

Acknowledgements

The authors gratefully acknowledge the funding from (1) the National Research Foundation (NRF) Singapore under the Energy Innovation Research Programme (EIRP) Funding Scheme (R-265-00-543-279); (2) the National Research Foundation Singapore under its Campus for Research Excellence and Technological Enterprise (CREATE) programme; (3) the UK Engineering and Physical Sciences Research Council (EPSRC) Translational Energy Storage Diagnostics (TRENDs) project; and (4) the STFC Futures Early Career Award.

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