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Study on dew point evaporative cooling system with counter-flow configuration

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ABSTRACT

Dew point evaporative cooling has great potential as a disruptive process for sensible cooling of air below its entering wet bulb temperature. This paper presents an improved mathematical model for a single-stage dew point evaporative cooler in a counter-flow configuration. Longitudinal heat conduction and mass diffusion of the air streams, channel plate and water film, as well as the temperature difference between the plate and water film, are accounted for in the model. Predictions of the product air temperature are validated using three sets of experimental data within a discrepancy of 4%. The cooler's heat and mass transfer process is analyzed in terms of its cooling capacity intensity, water evaporation intensity, and overall heat transfer coefficient along the channel. Parametric studies are conducted at different geometric and operating conditions. For the conditions evaluated, the study reveals that (1) the saturation point of the working air occurs at a fixed point regardless of the inlet air conditions, and it is mainly influenced by the working air ratio and channel height; (2) the intensity of the water evaporation approaches a minimum at 0.2 to 0.3 m from the entrance; (3) the wet channel can be separated into two zones, and the overall heat transfer coefficient is above 100 W/(m^2·K) after the temperature of water film becomes higher than the working air temperature.

1. Introduction

Ever since Wills Carrier invented the electricity-driven refrigeration cycle in 1902 in Buffalo, NY, the overall cooling energy efficiency of chiller plants operating in the tropical regions has been asymptotic to about 0.85 ± 0.2 kW/Rton [1]. For buildings in the hot and humid climate, air conditioning systems account up to 50% of the building energy consumption [2,3]. For instance, in 2013, the annual share of energy consumption for cooling applications in the three key sectors of Singapore, namely residential, industrial and commercial, is 14.7 GW·h which is equivalent to about 33% of the total electricity consumption of 44.3 GW·h [1,4,5]. A major portion of this energy consumption can be attributed to electrically-driven mechanical vapor chillers and cooling towers. Such a high energy consumption rate is expected to increase with continued annual growth of 3–5% in both the GDP and the high rise buildings in Singapore. In view of the continued increase in electricity consumption for cooling, there is an imperative need for engineers and scientists to innovate cooling processes and technologies that can abate the increment.

Many attempts have been made to improve the performance of mechanical vapor compression refrigeration system. Refrigerant subcooling, two-stage systems, cooling control strategy, and different refrigerants have been investigated [6–9] while new cooling alternatives such as sorption and evaporative cooling technologies have been proposed and demonstrated in niche applications for improved efficiency [10–19]. Sorption systems (absorption and adsorption) can be driven by solar energy or waste heat [12,13]. However, in practice, such systems are rather complicated with relatively low coefficient of performance (COP). Besides, these technologies involve the transportation of the large amount of heating and cooling media and are only applicable where a high-quality heat source is abundant. On the other hand, evaporative cooling technologies utilize the latent heat of evaporation to cool down the air without the need for compressor and cooling tower, realizing low electricity consumption while achieving the essential cooling power. For a typical evaporative cooling device, the COP in terms of electrical power can be as high as 15–20 [14]. Furthermore, no chemical refrigerant is used in evaporative cooling systems and they are deemed as environmentally friendly. Hence,

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Nomenclature

A	area (m^2)
c	specific heat (kJ/(kg·K))
D_{AB}	mass diffusivity (m^2/s)
D_h	hydraulic diameter (m)
$\dot{E}$	energy storage (W)
Er	error (%)
h	specific enthalpy (kJ/kg)
$\bar{h}$	convective heat transfer coefficient (W/(m^2·K))
$\bar{h}'$	overall heat transfer coefficient (W/(m^2·K))
$\bar{h}_m$	convective mass transfer coefficient (m/s)
Gz_D	Graetz number
H	channel height (mm)
k	thermal conductivity (W/(m·K))
l	length (m)
Le	Lewis number
$\dot{m}$	mass flow rate (kg/s)
$\dot{M}$	species storage (kg/s)
n	mass transfer rate (kg/s)
Nu	Nusselt number
P	pressure (Pa)
Pr	Prandtl number
q	heat transfer rate (W)
$\dot{Q}$	cooling capacity (W)
$\dot{Q}_A$	intensity of cooling capacity (W/m^2)
r	working air ratio
Re	Reynolds number
t	Celsius temperature (°C)
T	thermodynamic temperature (K)
v	velocity (m/s)
V	volume (m^3)
$\dot{V}$	volume flow rate (m^3/s)
W	channel width (mm)
$\dot{W}$	water evaporation rate (kg/s)
$\dot{W}_A$	intensity of water evaporation (kg/(m^2·s))

Greek symbols

α	thermal diffusivity (m^2/s)
δ	thickness (mm)
ε	effectiveness
ρ	density (kg/m^3)
ω	humidity ratio (g/kg dry air)
Φ	relative humidity (%)

Subscripts

a	air
A	area
CV	control volume
d	dry
dp	dew point
f	water film
in	inlet
out	outlet
p	product
pl	plate
s	saturation
th	thermal
v	water vapor
w	wet
wb	wet bulb

evaporative cooling technology has gained much attention in air conditioning application.

In this paper, we focus on the dew point evaporative cooling process that will eventually result in a potential technology for air cooling and yet without the use of any refrigerant-based compressor cycle. Although the concept of evaporative cooling is not new, it can be categorized into two groups, namely, direct evaporative cooling (DEC) and indirect evaporative cooling (IEC). The former method entails cooling by humidity addition to the supply air whilst the latter method, if properly configured, allows cooling process to reach the wet bulb temperature of the entry air [20–22]. Recently, dew point evaporative cooling [22,23] has demonstrated better potential for an IEC with higher wet bulb effectiveness than 1.0. Two viable flow configurations, i.e., cross-flow and counter-flow arrangements, were adopted for dew point evaporative cooling process [24,25].

Hsu et al. [26] investigated three types of wet-surface heat exchangers, including unidirectional flow, counter-flow and closed-loop flow configurations. They reported that the maximum wet bulb effectiveness for counter-flow, cross-flow, and closed-loop configurations is 1.3. Zhao et al. [20] carried out a numerical study on a novel counter-flow heat and mass exchanger (HMX) with triangular air channels. They concluded that under a typical UK summer weather condition, the system could achieve the wet bulb effectiveness of up to 1.3. The effects of different parameters, such as air velocity, working-to-intake-air ratio and channel size on the cooling effectiveness were also studied. Duan [21] studied the dew point indirect evaporative cooler for building applications. The cooler was simulated and tested under different controlled parameters. It was found that the wet bulb and dew point effectiveness varied from 0.55 to 1.10 and 0.40 to 0.85, respectively, with COP ranging from 3 to 12. Riangvilaikul and Kumar [27,28] performed numerical and experimental studies on dew point evaporative cooling using different inlet air temperature, humidity and velocity, and the wet bulb effectiveness was found to be in the range of 0.92–1.14. Their proposed design performed well under dynamic conditions. Lee and Lee [29] proposed a counter-flow regenerative evaporative cooler with finned channels, and their experiment showed that under the inlet condition of 32 °C and 50% RH, the outlet temperature was 22 °C, below the inlet wet bulb temperature of 23.7 °C. Zhan et al. [30] compared the counter-flow and cross-flow dew-point cooling configurations, and evaluated their effectiveness, COP, and cooling capacity. From their simulation results, the counter-flow configuration provides better effectiveness and larger cooling capacity without significant increase in energy consumption, compared to cross-flow system. Jradi and Riffat [31] presented a two-dimensional numerical model for a cross-flow dew point evaporative cooler, and results were validated by the experimental data from the cooler. They stated that the wet bulb effectiveness of the cooler was 1.12 with 2017 W of cooling capacity under the inlet air condition of 30 °C and 50% RH. Hasan [32,33] proposed an analytical model using modified ε-NTU method, as well as the numerical method, to study the counter-flow dew point evaporative cooler. It was found that the results from the two models were similar, and agreed well with the experimental data. Anisimov et al. [24,34–37] and Pandelidis et al. [25,38–40] also presented a modified ε-NTU method for different configurations of M-cycle HMXes. Their model accounted for the fin surface performance and air mixing process in the channels, and addressed the detailed